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NAVY SWITCHBOARD INSTRUMENTS.

BY LT. COMDR. C. S. GILLETTE,* U. S. NAVY, MEMBER.

The increase in the number of electrically driven battleships built and building has brought many new problems for our designing and operating engineers to solve—not the least of which is the problem of properly metering and indicating the various electrical quantities involved in the functioning of the propulsion circuits.

The operator in the isolated control room neither sees nor hears the operation of any of the powerful machinery which he controls. He is in touch with it directly only through the large group of instruments on the board before him and by these he must judge not only the proper response to his switching operations, but the efficient and economical conditions as

*Material assistance was received from Mr. Clarence Huey, Chief Assistant Electrical Inspector of the New York Yard Laboratory in the preparation of sketches, description, etc.

well. Of course he has many systems of communication with the personnel in direct charge of the machinery units in any one of the many separate compartments under his control, but these cannot tell him so quickly nor so accurately as the silent shifting of one or the other of his instrument pointers a fraction of an inch or so.

It seems that it might be of interest, therefore, to discuss some of the general characteristics of the instruments used and bring out, if possible, wherein their application to electric ship propulsion differs from their use on ordinary land service.

The forms and types of electrical instruments are so many and varied that only those instruments which may be found in the control rooms of electrical propulsion ships will be discussed. Further, as instruments for use with direct current are more or less familiar to all, particular attention will be given to those instruments and accessories used with alternating current.

Of these the ordinary ones, *viz.*, ammeters, voltmeters, wattmeters, frequency meters, power factor meters and watthour meters are used on the main propulsion board and, in addition, meters for special purposes, such as the so-called "stability meters" and "temperature indicating meters." Most of these instruments, owing to the comparatively high voltages and heavy currents used for propulsion purposes, are operated on the secondaries of either current or voltage transformers, or both.

The instrument transformers are similar in principle to lighting or power transformers. Due to the very small power required to operate one or several instruments, the transformers are of small proportions and still air cooling is all that is required to maintain safe temperatures of the windings. The voltage transformers have been standardized to step-down from the normal plant voltage to a normal of 110 volts at the instrument, while the current transformers step-down from normal plant current to 5 ampères normal on the instrument.

With regard to the actual meters, the operating principles

are the same as for commercial types. However, the commercial types that can be successfully utilized for ship propulsion are limited by the peculiarities of the service required. So also the construction of the meters must be of a higher class to function under more severe conditions than commercial meters are ordinarily called upon to withstand.

By far the most important of the peculiar features of ship propulsion as compared to commercial usage, arises from the variation of the voltage and frequency over wide ranges. The practice for which commercial types of meters are designed, contemplates changes in the voltage and frequency over comparatively narrow limits. In obtaining speed variation of the ship by direct change in the R.P.M. of the turbo alternator, the frequency and voltage are naturally very widely varied as the frequency or alternations are directly proportional to the speed of the alternator and the voltage at the motors is practically proportional. Thus all meters operating on a principle not inherently independent of changes in these two quantities are excluded from consideration. Of course, this does not include meters designed to indicate one or the other of these same two quantities. In this case an instrument designed to indicate one must be independent of variation in the other and vice versa.

It is true that suitable calibration of the scale for any individual installation could be accomplished and this has been done; the two quantities, *viz.*, voltage and frequency, are closely tied together for any particular installation. But this cannot cover all conditions and is therefore a makeshift, to say nothing of the difficulties connected with a replacement.

The next important point where it is believed ship installation differs materially from land practice is in amount of stray field to which the instruments mounted on the boards of our control rooms are exposed. The maximum effect due to stray field on an alternating current meter is produced by alternating currents of the same or nearly the same frequency as that of the current through the instrument and approximately in phase with it. The proximity of the meters to conductors carrying

heavy alternating currents, enclosed in a steel control room, mounted on heavy steel members of the switching structure and possibly, in some cases, the meters themselves mounted or recessed in steel panel boards, must of necessity produce stray fields of a magnitude unusual in commercial installations. Add to this the fact that the meters themselves are placed very closely together, owing to the limited space available, and must therefore have considerable effect on each other, and we have a problem necessitating very careful consideration of the method used in shielding. No figures are available on the exact amount of this stray field or of the amount of error likely to be expected from this cause on meters with shielding of the ordinary commercial type. Meters are to be tested with an alternating field of 5 gausses, with a limiting error of 3 per cent full scale value. 200 ampère turns will give a field of 5 C.G.S. units at center of a loop 50 c.m. in diameter.

The remaining points of difference are mechanical effects produced on the instruments, due to the unusually high operating temperatures our meters are likely to encounter and the effect of vibrations. With ordinarily well designed commercial type meters on which these effects have been well guarded against, the unfavorable conditions can be met to a great extent by close attention to the installation, recognizing that serious errors are likely to be caused by abnormal conditions of either the one or the other. During installation or later during repair periods careless workmen can easily ruin a full set of instruments by excessive hammering or pounding in their vicinity or on the structure which supports them. Likewise if the greatest care is not taken, removal of an instrument for repair or cleaning will ruin its accuracy if repaired by other than a competent expert or if carelessly handled during removal, transportation or replacement.

In our later ships, vibrations in the control rooms are not likely to be of very great importance as far as those caused by the operation of the machinery is concerned. The shock of gun fire, however, though infrequent, may be of such magni-

tude as to seriously derange the instruments if special care is not taken with the mounting. Even at the best, this effect will be serious. Vibration or shock will tend to destroy the balance and introduce friction in the bearings of the moving element. The moving elements of the majority of these instruments are supported on finely ground hardened pivots resting in cupped sapphire or diamond jewels. In a properly constructed instrument with friction at these bearings reduced to a suitable point the fineness of the pivoting results in very high bearing pressures figured in pounds per square inch, even though the weight of the moving element is but a few grams. Hence the serious results of shock in tending to increase the bearing surface and thus the friction load.

Temperature affects the instruments by changing the resistance of the copper in the circuits, resulting in variation in the current flowing and decrease or increase in the torque. In spring controlled instruments, high temperatures tend to weaken the springs. These two effects counteract each other and the resulting error is less from temperature effect than would appear at first thought. High temperature increases resistance, reducing torque, and at same time weakens the springs which resist the torque. Tests indicate that, in practically all instruments, effect of high temperature (150 degrees F.) results in relatively large temporary error while instrument is subjected to such temperature, and a somewhat less but still important permanent error when instrument is returned to normal temperatures.

A repetition of the test on the same instrument varies the amount of the first error to an appreciable extent, not consistent with any fixed rule apparently, but has little effect on the second error. In general it can be stated that instruments should not be subjected to temperatures in excess of 100 degrees F. without probability of introducing considerable permanent error when normal temperatures are resumed.

The general remarks preceding concerning accuracy do not cover all of the sources of error likely to be encountered but

only those from which it is believed more trouble is likely to be encountered in marine installations of electrical propulsive machinery than in usual commercial practice. Inasmuch as all instruments for marine propulsion work will be operated from instrument transformers, the errors resulting from the variation in the transformation ratios and phase displacements of the latter will of course affect the combined error of the former due to the many other causes.

A list of the many sources of error likely to affect an installed instrument follows. The complete list does not apply to all instruments. Certain instruments by their operating principles are inherently free from errors due to certain ones of the causes enumerated. Others by their construction are more or less protected against certain effects. But, in general, for any given set of instruments for electric propulsion work, a large majority will be affected by practically all of the sources of error listed.

A—Variation of frequency.
B—Variation of voltage.
C—Variation of temperature (external and internal).
D—Vibration and shock, resulting in increased friction if not more serious damage.
E—Stray field (A.C. or D.C.).
F—Moisture inside case resulting in corrosion of pivots with increased friction, or grounding, etc.

NOTE.—For all the above, marine requirements are considered more severe than land requirements.

Those that follow are the same for both:
G—Dust and dirt inside case obstructing air gap, pivots, etc.
H—Overshooting (high overloads).
I—Lack of proper mechanical balance of the moving element.
J—Bending of pointer, or scale displacement or warping.
K—Use of zero adjuster with non-uniform scale.

L—Variation in transformer ratio.

M—Phase displacement of transformers.

N—Electrostatic effects, caused by wiping off glass faces, etc.

O—Abnormal wave forms in alternating circuits.

P—Variation in power factor.

The algebraic sum of the various errors from the many causes produce the combination error of the instrument. While certain errors counteract others, the combination is likely to be a very considerable amount, and it can easily be seen how extremely important it is to eliminate or reduce to a minimum every one of the many effects producing them.

When the above long list of possible causes for inaccuracy in the readings of switchboard instruments is considered, it seems at first thought that it would be almost impossible to obtain from such an instrument any exact information as to the magnitude of the electrical quantities involved. It is a tribute to the ingenuity and genius of the designers of these instruments that they can be and are made sufficiently accurate for all practical purposes. Of course, many of the errors resulting from the different causes counteract each other, so that the final accuracy is much better than might ordinarily be expected.

The greatest difficulty arises from the fact that usually extreme accuracy in an instrument of this type is accompanied by lack of ruggedness in construction and is more expensive to produce. Hence for any given practical application it must be definitely ascertained just what degree of accuracy is necessary and a compromise reached between accuracy and ruggedness, and between economy, consistent with the results required, and the expense of over specifying.

Accuracy is usually specified as a percentage of the full scale value. This is about as satisfactory a method as can be devised, particularly if this specified percentage of full scale value is varied for different sections of the scale. However, it is

evident that the higher the scale can be marked the greater will be the actual error allowed. Some manufacturers have a tendency to take advantage of this and extend the scale unduly beyond the working range, when their design of instrument permits. In such case an overload test using the highest scale marking as the normal will counteract this tendency.

Specifications for the Naval Service fix the allowed error in percentage of full scale values by test under many of the individual causes listed, *viz.*, varying frequency, voltage and power factor where applicable, heat, vibration and stray field. In addition, the final accuracy, a combination of initial error and the accumulative error from the many tests is specified.

Acceptance tests of all the various instruments delivered on contract are allowed a certain percentage of initial error. To this initial error must be added the allowed transformer error and the probable error due to stray field and room temperature in order to arrive at an estimate of the possible maximum error when a new instrument is installed for operation. It must be understood that these errors may counteract each other more or less, but it is possible that they may all be additive.

Again after some service a certain additional error may be expected from vibration, temperature, corrosion or wear.

Possible maximum errors under normal conditions and for any service combination of frequency, voltage, and power factor, of instruments which have passed naval specifications can be listed as follows for the various types. It must be recognized that it is highly improbable that any combination of errors will reach such a figure, but it is believed desirable to consider how much an instrument can be inaccurate and still have nothing radically wrong with the mechanism.

The following errors are given in percentages of full scale value unless otherwise noted. It should be understood that the figures for maximum possible error following are to be considered as applying only to instruments that have passed Navy Specifications and are in perfect condition and properly applied and connected.

Type of Instrument	Errors under any service combination of voltage, frequency, power factor, temperature, stray field, vibration, etc.		
	Possible Max. error installed	Possible Max. service error	Probable Max. error
Voltmeter (A. C.)	3 per cent	4 per cent	2 per cent
Ammeters (A. C.)	3 per cent	4 per cent	2 per cent
Wattmeters (A. C. single or polyphase)	4 per cent	5 per cent	3 per cent
Watthour meters (A. C. single polyphase)	6 per cent	8 per cent	4 per cent
Frequency meters	1 cycle	2 cycles	1 cycle
Power factor meter (single phase)	4 per cent in power factor reading 100	6 per cent in power factor reading 100	3 per cent in power factor reading 100
Power factor meter (polyphase)	3 per cent in power factor reading 100	5 per cent in power factor reading 100	2 per cent in power factor reading 100
D. C. Ammeter	2 per cent	5½ per cent	3 per cent
D. C. Voltmeter	2 per cent	3½ per cent	2 per cent
D. C. Wattmeters	3 per cent	5 per cent	5 per cent

The following general characteristics of design of instruments are considered worthy of note.

All calibrating resistances and resistor coils are invariably wound with a special resistance wire, usually nickel-chromium or manganin, which have practically a zero temperature coefficient so that the ohmic resistance remains constant at all operating temperatures.

Field coils and coils of the moving element are usually wound with copper wire so that maximum conductivity can be obtained with minimum weight and space. The effect of temperature on the copper is offset by a high proportion of fixed resistance of zero temperature coefficient in series therewith. Aluminum wire is often used on moving coils of A.C. instruments to decrease weight of coil.

Whenever two controlling springs are used on moving coils they are usually placed one on top and one on the bottom and

the spirals wound in opposition to compensate for the inequalities of the individual springs. Phosphor bronze is used for springs because of its non-corrodibility, non-magnetic and fair conductivity characteristics. On moving coil instruments the springs are used to carry current to and from the coils.

All reactance coils have laminated iron cores so as to reduce the eddy current losses and heating. Non-inductive or straight resistance coils are wound on forms having no iron and the windings are usually put on non-inductively.

Nearly all of the instruments which may be found on the main propulsion switchboard of an A.C. installation are operated on the secondaries of instrument transformers. Without these appliances it would be difficult, if not impossible, to measure currents or voltages of large magnitude. It would be necessary to bring the leads from the main power circuits directly to the instruments with resultant danger to operators due to close proximity to high voltages.

By means of transformers to reproduce on a smaller scale and in a low voltage circuit, the variable quantity it is desired to measure or utilize in measurement, the use of standard low capacity switchboard instruments is permitted for all circuits.

The laws which govern instrument transformers are the same as those which in general govern power transformers. The construction of the former is special only in that they must be supplied in various shapes and of the smallest size suitable for the installation of which they are a part.

The instrument transformers are of a dry type and are of two general classes, *viz.*, "voltage transformers" and "current transformers." In the voltage transformer the ratio of the primary to the secondary voltage is the essential feature. The current transformer is a special development of the transformer principle and the ratio of the primary to the secondary current is the important consideration.

The voltage transformer is connected with its primary across the line while the current transformer is connected in series with the line. In the first, the flux is dependent on primary

current which varies as the line voltage and the secondary current is thus independent of line current and follows the line voltage. In the second, the case is different; the load on the secondary, or the impedance, is a fixed load consisting of instruments and the flux varies directly with the main line current, the primary being in series with same. Thus the secondary current is independent of the line voltage and follows the line current. The secondary voltage in the voltage transformer has a fixed ratio to the line voltage. In the current transformer the secondary voltage tends to increase or decrease sufficiently to force the current through the secondary to the instruments.

The primary current in all cases consists of two parts—one the "working" current required to balance the secondary current and the other an exciting current required to set up in the core a flux sufficient to generate in the secondary a voltage that will circulate the secondary current.

Perhaps a comparison of the short circuited and open circuited secondary conditions in each case would clarify this point. If the secondary of a voltage transformer is shorted, the current will increase in both the primary and secondary windings and sufficient heat will be generated to cause a breakdown, because the lowered resistance of the secondary allows high current to flow under influence of the secondary voltage which always bears a fixed ratio to the line voltage. The counter E.M.F. of the primary is also cut down by the high secondary current and the lowered impedance allows undue current in the primary. If, on the contrary, the secondary circuit is broken under load, no current except the small exciting current in the primary would flow and the voltage would still remain that of the fixed ratio.

Now if the secondary of the current transformer is short circuited, the only current that could flow would be that of the fixed ratio to the line current and the voltage would drop to just sufficient to overcome the impedance of the short circuit conditions. But if the secondary be broken under load, all the primary current, which is actually the line current, will become

exciting current, since there is no secondary current to be balanced. This would result in a very high flux which would highly saturate the iron and the secondary voltage would build up to such an extent as to endanger life and possibly puncture the insulation. The transformer would thus be permanently magnetized and its accuracy reduced. In such case it would have to be de-magnetized, which could be done by passing full load current through the secondary and gradually reducing the current to zero.

Voltage transformers are used on voltmeters, wattmeters, watthour meters, power factor meters, frequency meters and stability meters, as well as on relays and other applications. Current transformers are used on ammeters, wattmeters, power factor meters, watthour meters and stability meters, as well as on relays, etc.

In connecting up instruments of multiple connections, it is necessary to know the relative instantaneous direction of currents in the leads. Manufacturers indicate this by the use of corresponding letters or polarity signs on the various terminals.

The secondaries of all instrument transformers must be grounded as a precaution against danger from high voltage in case of punctured insulation between primary and secondary and to free the instruments from errors due to electrostatic charges.

There are two inherent sources of error in the design, *viz.*, ratio error and phase displacement error. Curves of variations due to these errors are usually furnished and for any given design are relatively constant.

The ratio above referred to is the "ratio of transformation" and this, with voltage transformers, is the quotient of the voltage of the primary divided by the voltage of the secondary. It is a variable quantity and though nearly equal to the quotient of primary turns divided by the secondary turns, it must not be confused with such ratio which is only an approximation. In the case of current transformers the above also applies except that the "ratio of transformation" is the quotient of the primary current by the secondary current.

By phase displacement is meant the phase angle of the transformer which is the small angle by which the voltage of the secondary or the current of the secondary differs in phase relationship from the theoretical 180 degrees opposition to the phase of the primary voltage or primary current respectively. In an ideal voltage or current transformer having no losses, the secondary voltage or current would be in exact opposition to primary voltage or current.

This is not actually the case and under most conditions the phase angle is leading the 180 degrees position. Increasing secondary load usually decreases this lead and if mainly non-inductive may cause variation in phase angle through the 180 degrees point to the lagging position.

Both these quantities, *viz.*, ratio of transformation and phase angle, are varied by changes in secondary or instrument load. In the case of voltage transformers this is taken care of approximately by rating them considerably lower than the designed output which could be safely carried. Further, the winding is such that the instruments give their correct ratio when carrying an average secondary load which is considerably less than the possible output or even the rated output. Thus, over the range from no load to rated load, the error, due to variations in the ratio, is relatively small. They have their maximum accuracy, however, at about half-rated load.

By the expression "instrument load" or "secondary load" is meant the power required to operate the instruments. Increasing load means adding more instruments or in other words, increasing the volt-ampère demand on the secondary. This is often known as the "burden" to differentiate between the regular power load and the load on the transformer.

The design of the transformers is such that their accuracy is not appreciably affected by service. In general it has been stated that it is desirable to check the calibration after about five years' service, provided they have not been subjected to overloads, primary short circuits, or open circuiting of the secondary of the current transformer under load.

In general, it may be stated that the "through type," having hinged or split cores for use in conductors without cutting them, should be calibrated with the instrument with which used. Other types having larger value for the full-load ampère turns may be calibrated separately.

Acceptance accuracy under present Naval specifications requires that the ratio error shall not exceed 1 per cent under any service combination of voltage, frequency or instrument "burden," and that the phase displacement error shall not be excessive.

Transformers of either the voltage or current types are invariably wound with copper wire. Although copper has a high temperature coefficient, the ratio of transformation is not affected to any extent by self-heating or changes in room temperature.

In using voltage transformers, two, three or even four voltage coils of as many instruments can be placed in parallel on the secondary without materially affecting the accuracy, providing the load is well within the rated volt-ampère capacity of the transformer. Thus, the same transformer can be used if convenient for the voltage coils of a voltmeter, a frequency meter, a wattmeter and often a power factor meter all connected in parallel.

With current transformers the same is true, except that all current coils are connected in series with the secondary. Thus, if well within the volt-ampère capacity, the current coils of an ammeter, wattmeter and power factor meter can be connected in series on the same transformer.

The secondaries of voltage transformers can be left open circuited indefinitely. But the secondary of a current transformer can never be open circuited for an instant without danger of ruining the transformer and possibly of injury to personnel.

Instruments require no oiling and should never be opened up for any purpose except by an expert instrument man. At present Naval instruments are furnished "moisture and dust

proof," and it is the present intention to furnish future instruments sufficiently watertight to withstand three-foot submergence, in order to insure this condition. In operation care should be taken to prevent high overloads, high temperature, rough handling, and vibration.

Instruments should be mounted with due regard to the influence of any possible stray fields and their influence one upon another. Protection against undue vibration or shock should be provided where possible and exposure to abnormally high temperatures or wide temperature changes studiously avoided. Particularly should care be taken to prevent all hammering, pounding or drilling on the switchboard structures or supporting bulkheads in their vicinity. If repair work necessitates such procedure, the instruments should be taken down carefully and stowed away until such work is completed, care being taken at the same time so that no mistake can be made when reconnecting the leads. All contacts should be carefully made and kept clean and tight. Special means should be employed to see that contacts remain tight under service conditions. It must also be borne in mind that instruments are not of much use unless they can be easily read, and therefore they should be installed with due regard to lighting and at a point most convenient for the operating personnel.

Special attention should be given to ammeters having separate shunts. The Navy has standardized on 50 millivolt ammeter shunts. It is important that ammeter shunt connections receive the greatest care to eliminate any resistance at these points, also that the leads from the shunt to the instrument are always considered a part of the instrument, as they are specially calibrated for use with it. Delivery of the leads with the instrument should be insisted upon and, in most cases, the shunt also, if it is a separate shunt instrument.

It is considered good practice to check important instruments where possible with the secondary standard or portable instrument, whenever available for the purpose, at least once each month, or more often, if there is reason to suspect inaccuracy.

The portable secondary standards should themselves be checked with primary standards at shore laboratories whenever opportunity offers, every month or so if possible in the case of important instruments which are frequently in use. Resistance standards should be checked occasionally, every year or so at least. Watthour meters should be checked up once every two weeks if possible.

In general, no attempt should be made to repair or adjust instruments on board ship. The zero adjustment is about the only one that should be attempted by any one except an experienced instrument man. Nearly all voltmeters, ammeters, and wattmeters have a zero adjuster accessible from the outside of the case which can be used to bring the pointer to a zero reading when instrument is not in use. The range of the zero adjustment is small but is sufficient to take care of any zero shift due to normal operation. It should not be used to correct for a bent pointer. The mechanical balance can be checked without opening the case by turning the instrument in different positions around some axis parallel to the axis of rotation of the element and noting the change of the zero reading. It can only be corrected, however, by opening the case and shifting the balance weights.

A.C. voltmeters and ammeters, except the induction type, can be checked with sufficient accuracy on direct current, using a D.C. portable secondary standard. A convenient test circuit can be made up by using a variable lamp load for testing throughout the scale. It should be noted that most A.C. instruments are operated from instrument transformers and, though the scales are marked with the line load, the instrument circuits are functioning under a normal of 110 volts and 5 ampères as a rule. These quantities should always be verified by ascertaining the exact transformation ratio for any given installation.

Wattmeters are calibrated to show the effective power, and in service are often single phase with scales marked for total power, and can be checked as installed by the readings of the

voltmeter, ammeter and power factor meter on the line with it, having due regard to the phases of the circuit and the number of meters installed. Usually a balanced load can be assumed on all vessels. Frequency meters can also be readily checked as installed, by comparing the reading with the true frequency determined from the speed of the alternator. The same applies to power factor meters working back from the readings of wattmeter, voltmeter and ammeter.

D.C. watthour meters can be checked roughly to determine serious installation errors by determination of the average voltmeter and ammeter readings over a time interval when the load is fairly steady. On an A.C. system, the average wattmeter reading can be used over an interval of time. For more accurate checking they would have to be taken from the board and hooked up in a test circuit with suitable lamp load. The revolutions of the disc should be counted with an accurate stop watch for a period of about one minute and the load held steady and read by test ammeter and voltmeter in the circuit. For A.C. work, as may be noted, a lamp load would be satisfactorily non-inductive. The constant of the watthour meter will be found marked on the name plate and shows the watthours corresponding to each revolution of the disc. The calculation is then made using the disc R.P.M., the time interval and the constant of the instrument.

It is of special importance with all instruments and transformers to see that the electrical connections are exceptionally well made. In all cases of suspected inaccuracy, it is well to check the zero adjustment first, then inspect the scale for warping or possible interference due to a bent pointer, and finally check up all the connections before anything further is done.

It is desirable to know in cases of suspected inaccuracy just what the error of any given instrument may be. The means available for this purpose on a Naval vessel are very limited. In the majority of cases this also should be done by an expert instrument man, but it is believed that it would be of value to

point out for the different instruments what, if any, are the means available on the ordinary vessel of the Navy to accomplish this result.

The first step is, of course, to ascertain the working principle of the instrument to be dealt with, which is not always as easy as it sounds. Some foreign manufacturers mark their instruments with a symbol indicating the working principle, but this is not American practice and so very often the instrument must be opened up to find out this most important point.

In the case of voltmeters and ammeters, we may have some one of the following on a Naval vessel:

1. D.C. voltmeters and ammeters permanent magnet moving coil type.
2. A.C. voltmeters and ammeters dynamometer moving coil type.
3. A.C. voltmeters and ammeters electromagnetic or moving iron type.
4. A.C. voltmeters and ammeters induction type.

The latter type, being inherently unsuitable for use with variable frequency, will not be found on electrically propelled vessels except the U. S. S. *Tennessee.*

The above types of voltmeters and ammeters, excepting class 4, may all be calibrated with sufficient accuracy on direct current, which on shipboard is usually the most available steady source of supply. Current may be taken from the D.C. generators or preferably from some group of storage batteries of sufficient total voltage or ampèrage to give the maximum desired reading. The most satisfactory means of checking is to make a direct comparison with a portable direct current shielded type instrument similar to Weston model No. 45. These meters are reliable if not abused and accurate to 1/2 of one per cent. Verification of the accuracy of the portable can be made, and should be whenever convenient, by reference to the nearest laboratory. Class 1 instruments should be connected with correct polarity or else they will read backwards; classes 2 and 3 will always read in the right direction regard-

less of polarity. The average of two readings of reversed connections should be taken to offset the residual magnetic effect of the iron shield used in class 2 and the iron moving coil and shield used in class 3. Class 4 can be calibrated only on A.C. current and for this purpose an accurate portable A.C. instrument should be used, preferably of the shielded dynamometer type.

For checking the class 1 meters the volts or ampères marked on the scale invariably are the actual volts or ampères, or the ampères through the ammeter shunt, to be applied from the test circuit. This also applies to classes 2, 3 and 4 where no instrument transformers are installed. On high range meters where instrument transformers are used, the volts or ampères to give full scale deflection and the ratio of reduction of voltage or ampères through the transformer, are usually marked on the scale. Ordinarily on high reading voltmeters, 110 volts will give normal deflection on the voltmeters and 5 ampères on the ammeters. A convenient test circuit for calibrating may be arranged as follows:

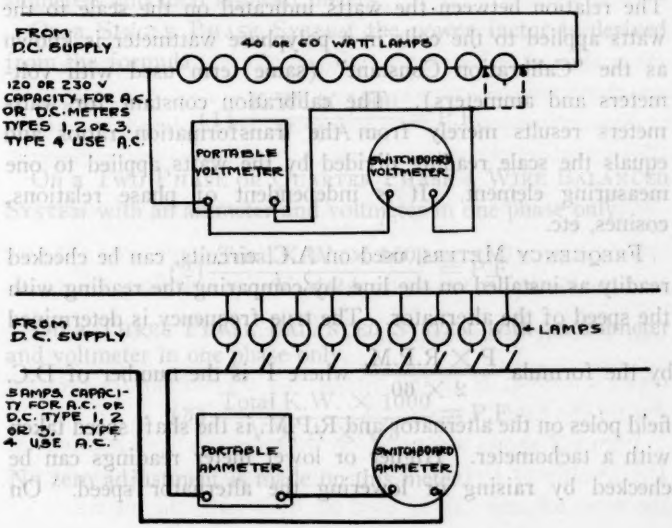

FROM D.C. SUPPLY
40 OR 60 WATT LAMPS
120 OR 230 V CAPACITY FOR A.C. OR D.C. METERS TYPES 1, 2 OR 3. TYPE 4 USE A.C.
PORTABLE VOLTMETER
SWITCHBOARD VOLTMETER
FROM D.C. SUPPLY
LAMPS
5 AMPS. CAPACITY FOR A.C. OR D.C. TYPE 1, 2 OR 3. TYPE 4 USE A.C.
PORTABLE AMMETER
SWITCHBOARD AMMETER

FIGURE 1.

In the case of wattmeters, Naval specifications limit the type to the electrodynamometer only.

Kilowattmeters have scales calibrated to show the total true electrical power and usually only one meter is used on one, two or three phase installations. The meter may be checked as installed by figuring out the true kilowatts derived by readings of the voltmeters, ammeters and power factor meters.

On a SINGLE PHASE SYSTEM the kilowatts power is derived from the formula

(1) Volts $\times$ ampères $\times$ P.F. = Watts.

On a TWO PHASE OR QUARTER PHASE 4 WIRE SYSTEM with an ammeter and voltmeter in one phase only, assuming balanced load

(2) $V \times 2A \times$ P.F. = Total Watts.

On a THREE PHASE SYSTEM with an ammeter and voltmeter in one phase only, assuming balanced load

(3) $V \times A \times \sqrt{3} \times$ P.F. = Total Watts.

The relation between the watts indicated on the scale to the watts applied to the coils in a polyphase wattmeter is known as the "Calibration Constant" (same term used with voltmeters and ammeters). The calibration constant for wattmeters results merely from the transformation ratios and equals the scale reading divided by the watts applied to one measuring element. It is independent of phase relations, cosines, etc.

FREQUENCY METERS, used on A.C. circuits, can be checked readily as installed on the line, by comparing the reading with the speed of the alternator. The true frequency is determined by the formula $\frac{P \times \text{R.P.M.}}{2 \times 60}$ where P is the number of D.C. field poles on the alternator and R.P.M. is the shaft speed taken with a tachometer. Higher or lower meter readings can be checked by raising or lowering the alternator speed. On

account of the small current taken by the meter (a fraction of an ampère) the meter can be disconnected from the board with its reactance box, if it has one, and checked on some small alternator of normal frequency within range of the meter. The meter can be connected across the full line volts in the same manner as a voltmeter. The voltage of the alternator will have no effect on the calibration, providing it is somewhere near the normal voltage marked on the meter, usually about 110 volts. The number of poles on an alternator can be determined ordinarily from the name plate, *e. g.*, 1,800 R.P.M., 60 cycles sets would have $\frac{60 \times 2 \times 60 \text{ cycles}}{1{,}800 \text{ R.P.M.}} = 4$ poles. No zero adjustment is made on this meter.

Power Factor Meters are installed usually with only one meter for one, two or three phase system. The meter reads the power factor in the two or three phases assuming balanced load. The meter may be checked as installed by figuring out the true power factor derived by readings of the voltmeters, ammeters and total kilowatt meters.

On a Single Phase System the power factor is derived from the formula

$$(1)\ \frac{\text{K.W.} \times 1000}{\text{V} \times \text{A}} = \text{P.F.}$$

On a Two Phase or Quarter Phase 4 Wire Balanced System with an ammeter and voltmeter in one phase only.

$$(2)\ \frac{\text{Total K.W.} \times 1000}{\text{V} \times 2\text{A}} = \text{P.F.}$$

On a Three Phase Balanced System with an ammeter and voltmeter in one phase only.

$$(3)\ \frac{\text{Total K.W.} \times 1000}{\text{V} \times \text{A} \times \sqrt{3}} = \text{P.F.}$$

No zero adjustment is made on this meter.

WATTHOUR METERS can be checked roughly to determine serious installation errors, by means of observations of the average readings of the ammeter and voltmeter on a D.C. system, over a period of time when the load is fairly steady. For example, if the volts are 125 and the current is 800 for a period of one hour, this can be compared to the dial reading of the watthour meter which should show an increase of 100 K.W. hours during the one hour period. On a three wire system the current in the two legs should be considered in figuring the total if the meter is a three wire meter. On an A.C. system, instead of reading the voltmeter and ammeter, read the kilowatt meter for an hour. This reading should compare direct with the kilowatt hour meter. For a more accurate check on watthour meters, they may be taken down from the board and calibrated by small current A.C. or D.C. as the case may be. The name plate will indicate the voltage and current to give the maximum watthour reading, usually 110 volts and 5 or 10 ampères. The method then is to count the revolutions of the disc on the meter by means of an accurate stop watch for a period of about one minute. At the same period hold a steady test load and read the volts and ampères for a D.C. or A.C. test as shown in figure 2. The calculations are then made as follows:

The actual watts are the volts $\times$ ampères of the test meters. The actual watthours are, watts $\times$ duration of test in seconds, divided by 3600 or $\frac{W \times T}{3600}$. The watthour meter under test will have a certain constant marked on the name plate showing how many watthours each revolution of the disc will equal. The reading then becomes $\frac{K \times R \times 3600}{T}$ where K is the meter constant, R is the number of revolutions of the disc during the time in seconds T.

Figure 2 shows a sample test arrangement for checking the D.C. watthour meters, and as the lamp load is non-inductive the same schemes can be used on A.C. watthour meters.

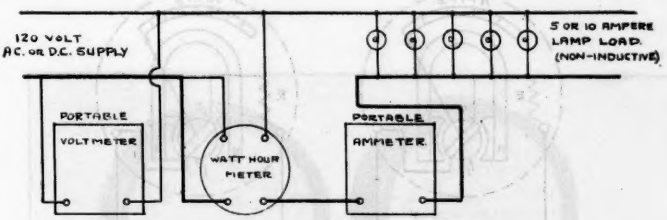


FIGURE 2.

Instrument transformers can only be checked in a laboratory having special equipment to perform this work.

Special instruments require special methods. The potentiometer for measurement of temperature in stator windings is checked by the standard cell furnished therewith as described hereafter. The "stability meter" and "temperature indicator" can be calibrated only by the manufacturer who has full access to complete data concerning performance of the motors and alternators for any given installation.

There follow descriptions and sketches showing operating principles and connections for several types of instruments that may be encountered. No attempt is made to cover all the types or connections in the application, as this would be almost impossible. But it is believed enough are given to make it possible to understand any combination which may be encountered in the Naval Service.

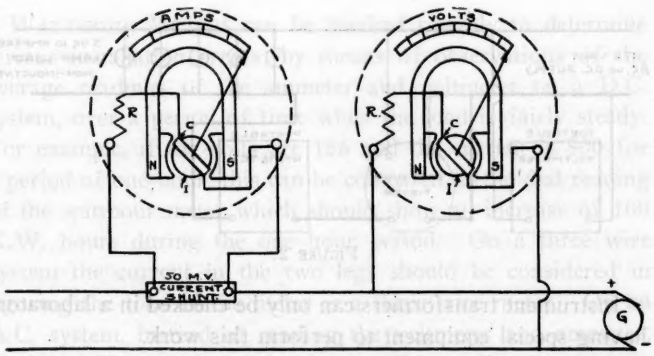


FIGURE 3.—D. C. METERS, PERMANENT MAGNET, MOVING COIL TYPE. Weston, General Electric, Westinghouse.

THEORY: A permanent magnet of the horseshoe form is employed to produce a magnetic field of constant flux across the air gap in which revolves the moving coil C. The moving coil is rectangular in shape and wound with copper wire which carries the current or a portion of the current to be metered. The current is carried in and out of the coil by two controlling springs. The coil surrounds a cylindrical soft iron core I, which is used to concentrate the field and to distribute the flux in the air gap so as to produce uniform graduations on the scale. When current flows through the moving coil a flux is produced at right angles to the coil. The reaction between this flux and the flux from the permanent magnet causes the coil, to which is attached the pointer, to take a resultant position which is calibrated and adjusted by means of fixed resistance R in the instrument.

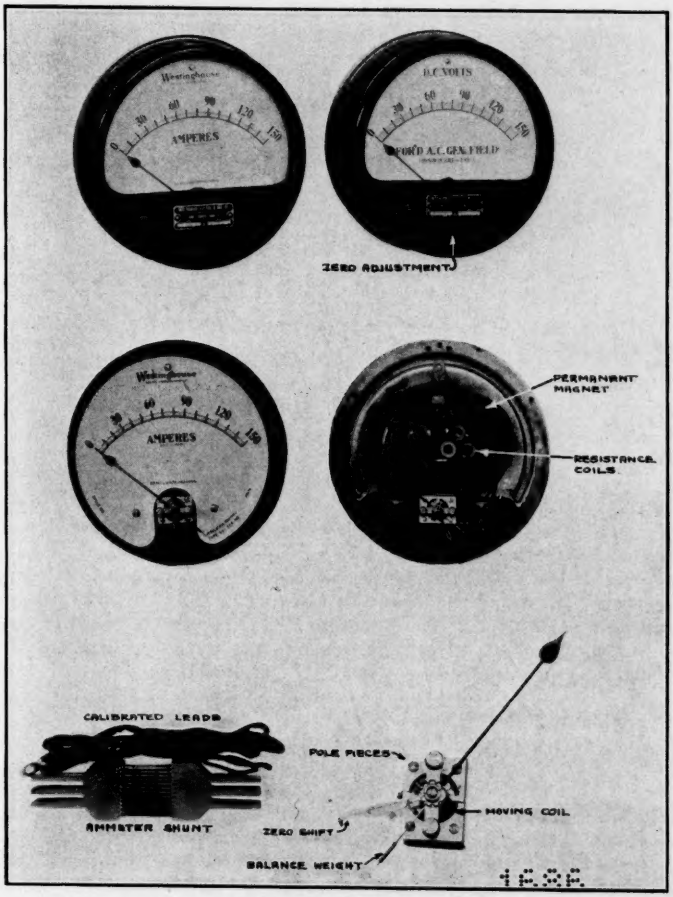

AMPERES
D.C. VOLTS
FORD A.C. GEN. FIELD
ZERO ADJUSTMENT
AMPERES
PERMANENT MAGNET
RESISTANCE COILS
CALIBRATED LEADS
AMMETER SHUNT
POLE PIECES
ZERO SHIFT
MOVING COIL
BALANCE WEIGHT

FIGURE 38.

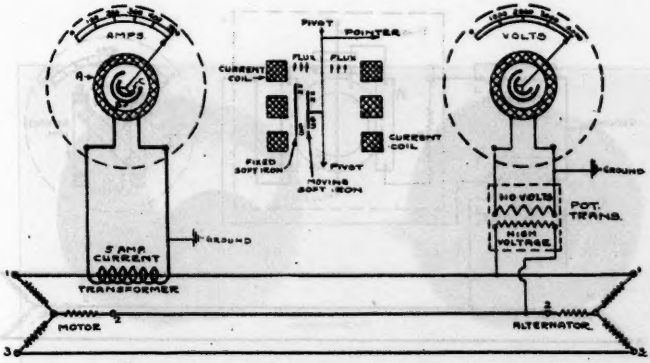


FIGURE 4.—AMMETER OR VOLTMETER, ELECTRO MAGNETIC OR MOVING IRON TYPE.

Westinghouse Type SY Ammeter, Weston Model 260, Showing Internal and External Connections.

THEORY: In the ammeter the moving element consists of a small curved piece of iron C fastened to a light pivoted shaft on which is mounted the pointer, air damping vane and controlling spring. Near this movable element and concentric with it is a small curved tongue of soft iron B, which is rigidly held by a suitable support. Surrounding these is a field coil made up of a few turns of heavy copper through which the load current passes. When A. C. current passes through the coil A, the movable element iron and the fixed tongue of iron become magnetized by similar polarities by virtue of their similar positions in the flux produced by the coil. By magnetic laws "like" poles repel each other. The only resultant motion possible is the rotation of the movable element controlled by the torsion spring on the shaft. As the alternating current reverses, the flux and the polarities reverse, but as the fixed iron and the moving iron are always "like" poles in relation to each other, they repel each other in continuous impulses causing the deflection of the moving element in one direction only and in amount depending upon the current in the coil. The success of the operation as described is largely due to the peculiar shapes as well as the relative positions of the iron parts which is the result of experiment. By design of the parts, the hysteresis effect of constant reversals of magnetization has been reduced to a neglible amount. The theory and application of the voltmeter is identical with the ammeter, except that the flux coil consists of many turns of fine wire in shunt across the power voltage. The amount of flux and the consequent deflection of the pointer varies with the impressed voltage and the adjustments are made to calibrate the scale to read volts.

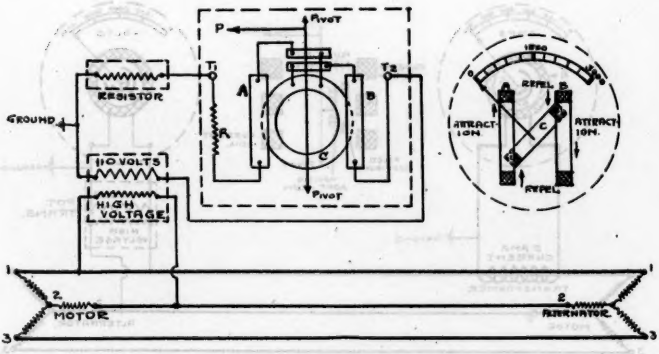


FIGURE 4 A.—A.C. VOLTMETER, DYNAMOMETER TYPE.
Westinghouse Type SY, Showing Internal and External Connections with Resistances and Transformers.

THEORY: The dynamometer principle depends on the phenomena that when current flows in the same direction in two parallel conductors they mutually attract, and conversely when current flows in opposite directions, they mutually repel. In the sketches which show the relation of the current carrying circuits in two planes, coils A and B are fixed, and coil C is the moving coil carrying the pointer. The coils are all in series and are wound in such a direction that the resultant operation of the mutual attraction and repulsion between the coils will be as shown in the cross-section sketch. A non-inductance coil R of high resistance and low temperature coefficient is placed in series with the fixed and movable coils to reduce the frequency and temperature errors to a neglible amount. When current flows in the circuit due to the impressed voltage on the terminals, the deflecting force varies as the square of the current. On account of the change in relative position of the coils when in use, exact proportionality does not exist between the deflecting force and the square of the current. Since the counterforce of the controlling springs varies directly with the deflection, dynamometer type instruments do not have uniform scales, the graduations being crowded together at the beginning and end of the scales. If the pointer is located off the center line of the movable coil, the crowding can be confined to the beginning of the scales and the upper end of the scale may be made fairly uniform or vice versa, Dynamometer type instruments operate equally well on D. C. or A. C. currents, as the coils are all in series and reversing of the current at the terminals does not change the resultant attraction and repulsion forces as the direction of the deflection is fixed by the designed direction of the windings on the fixed and moving coils.

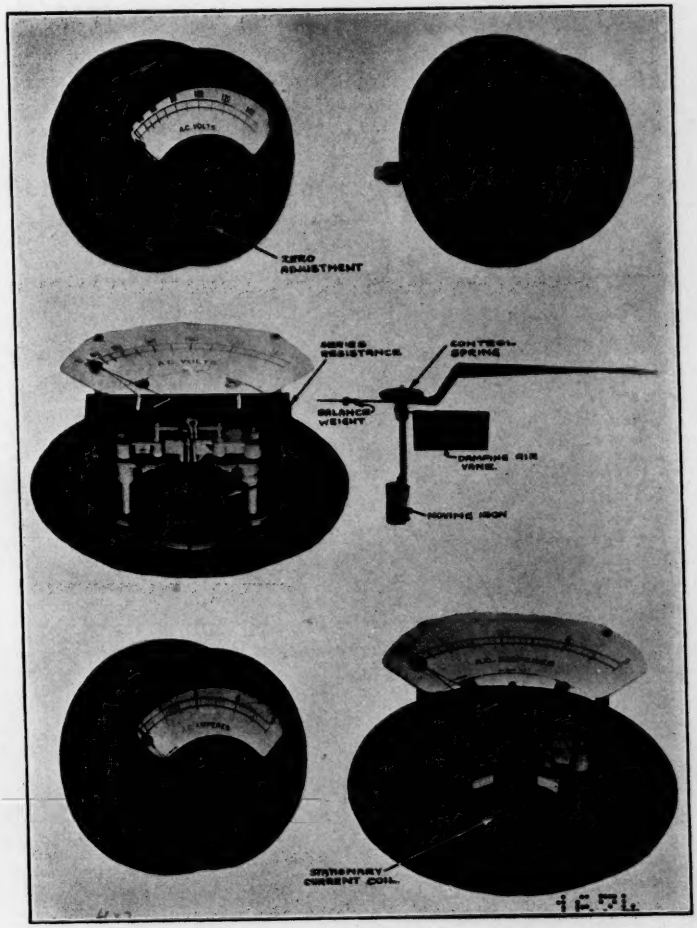

ZERO ADJUSTMENT
SERIES RESISTANCE
CONTROL SPRING
BALANCE WEIGHT
DAMPING AIR VANE
MOVING IRON
STATIONARY CURRENT COIL

FIGURE 4 a.

FIGURE 4 Aa.

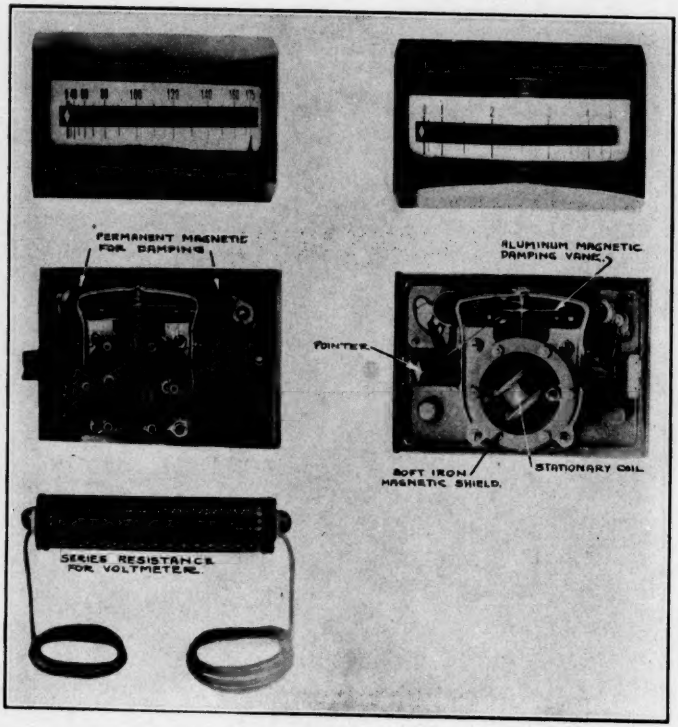

PERMANENT MAGNETIC FOR DAMPING
ALUMINUM MAGNETIC DAMPING VANE.
POINTER.
SOFT IRON MAGNETIC SHIELD.
STATIONARY COIL.
SERIES RESISTANCE FOR VOLTMETERS.

FIGURE 5a.

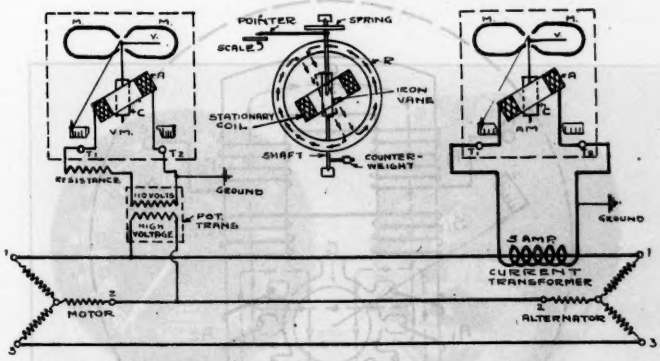


FIGURE 5.—VOLTMETER AND AMMETER, ELECTRO-MAGNETIC, MOVING IRON, INCLINED COIL TYPE.

General Electric Co. Type H2.

THEORY: The stationary coil A consisting of a few turns of heavy copper wire for the ammeter and a large number of turns of fine copper wire for the voltmeter is placed at about 45 degrees with the shaft upon which is mounted a soft iron vane C, also making an angle of 45 degrees to the shaft. Current flowing through the stationary coil due to the voltage or ampères of the power current, as the case may be, sets up a magnetic field as shown by the arrows in the sketch. The laminated soft iron ring R serves as a magnetic shield from external stray fields and also to increase the field intensity through the center of the stationary coil A. When the pointer is at zero of the scale, the plane of the vane is nearly at right angles to the magnetic lines. When the magnetic field is set up, the vane moves so as to place itself parallel to the magnetic field. Opposing the motion is the controlling spring S at the upper end of the shaft. The pointer being rigidly attached to the shaft moves until the torque due to the controlling spring balances the magnetic pull. Damping of the moving element is accomplished by means of an aluminum sector fixed to the shaft and which revolves between the poles of a strong permanent magnet. Eddy currents set up in the sector as it moves through the field of the permanent magnet resists the movement of the sector.

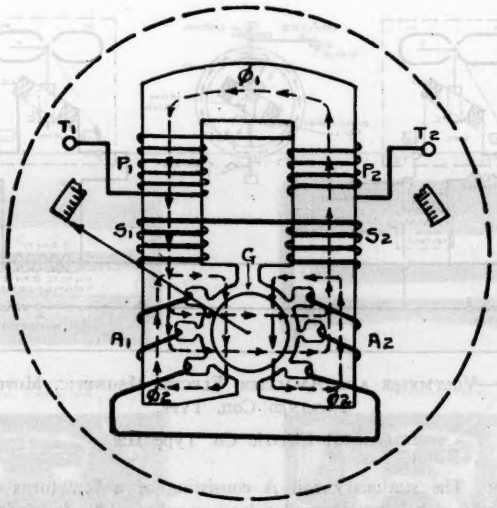


FIGURE 6.—A. C. AMMETER OR VOLTMETER INDUCTION TYPE.

Westinghouse Model SM, Showing Internal Windings and Connections.

THEORY: The induction type of meter consists of a scheme of transformer principle to produce a rotating field in the air gap wherein is located the moving element consisting of a light aluminum cup G controlled by torsion springs. The current to be metered passes around a laminated iron core in coils P_1 and P_2 in similar arrangement to the primary of a transformer. A secondary coil system S_1 and S_2, by induction from the flux set up by the primary causes current to flow around a closed circuit consisting of auxiliary coils A_1 and A_2. The flux ϕ_1 produced as a combined effect of both the primary and secondary coils passes across the air gap of the moving coil system in alternating directions from left to right. The flux ϕ_2 produced by auxiliary coils, due to the mechanical arrangement and position of the pole pieces, passes across the air gap of the moving coil at right angles to flux ϕ_1. The flux ϕ_2 due to the inductance of the auxiliary coils and the element of design and experiment is caused to lag 90 degrees behind the flux ϕ_1. This lag in conjunction with the direction of the fluxes across the moving coil air gap produces a condition of a rotating magnetic field. This condition, in turn, due to eddy currents set up in the aluminum moving coil produces torque similar to an induction motor. The torque controlled by the counter-acting control springs is calibrated to read current in the primary circuit. The principle of the induction voltmeter is identical except that many turns of fine wire is used on the primary coils, and as the current is proportional to the voltage at a constant frequency, the meter is calibrated to read Volts.

NOTE.—This type of instrument has many excellent features, but as it is inherently very susceptible to changes of frequency, its accuracy is not considered sufficient for A. C. electrical propulsion circuits where frequency varies over a considerable range. It is used on the switchboards of the U. S. S. *Tennessee* only.

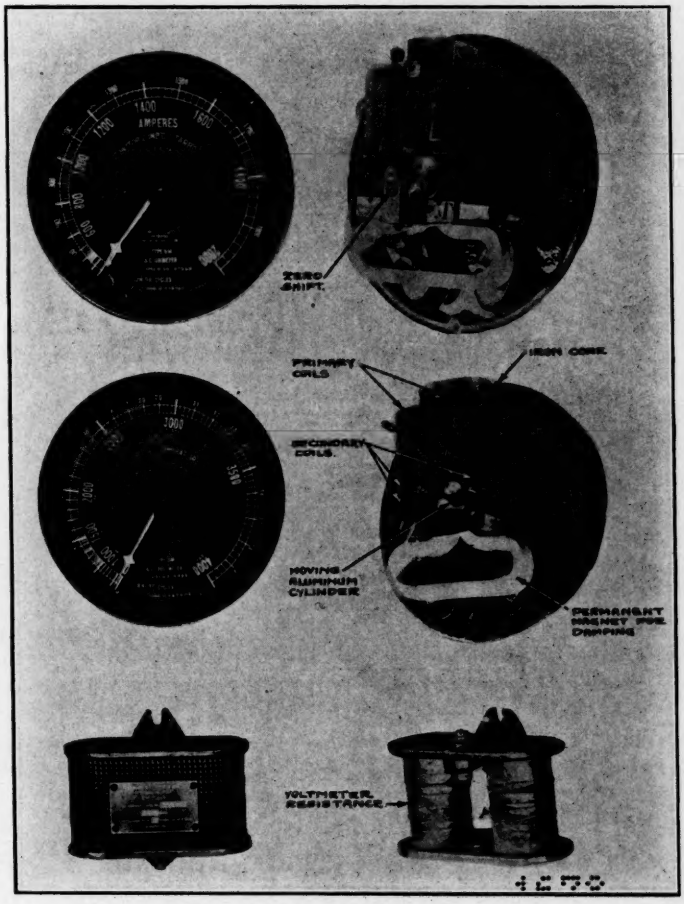

1400
AMPERES
1600
1200
800
400
ZERO SHIFT.
PRIMARY COILS
IRON CORE.
SECONDARY COILS.
MOVING ALUMINUM CYLINDER
PERMANENT MAGNET FOR DAMPING
VOLTMETER RESISTANCE

FIGURE 6a.

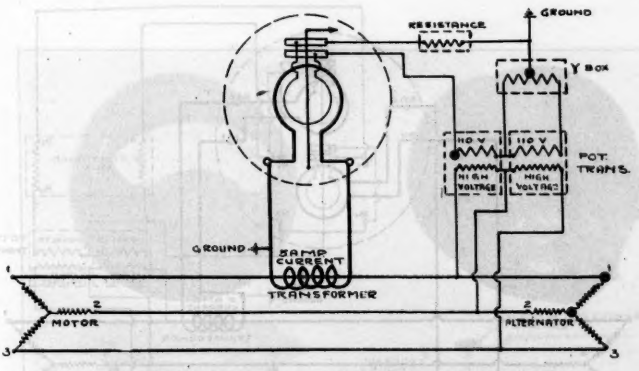


FIGURE 7.—WATTMETER SINGLE PHASE-DYNAMOMETER TYPE.

Westinghouse Model SY, Showing Internal and External Connections and Hook-up to 3 Phase 3 Wire System.

The single phase wattmeter or kilowatt meter is used, connected up as above, on three wire three phase systems where the load is assumed as being balanced at all times, as is the case with A.C. ship propulsion. Since the meter is to read the total power, the scale value is arbitrarily multiplied by the factor three and marked accordingly. The meter normally would read only the power in one phase. The Y box, consisting of a non-inductive resistance, is a device to furnish an accessible or artificial neutral point as shown by the small circles, so that the true armature voltage of one phase may be obtained for the voltage coil of the instrument. Line volts is not used directly as the line volts in the star connected generator is 1.73 times the voltage of one armature phase. This is not the case with the current, as the line current is the true armature current of each phase and can be used direct on the current coil.

The reactions, giving the indications, are the same in principle as covered under the polyphase wattmeter for individual elements, description of which follows.

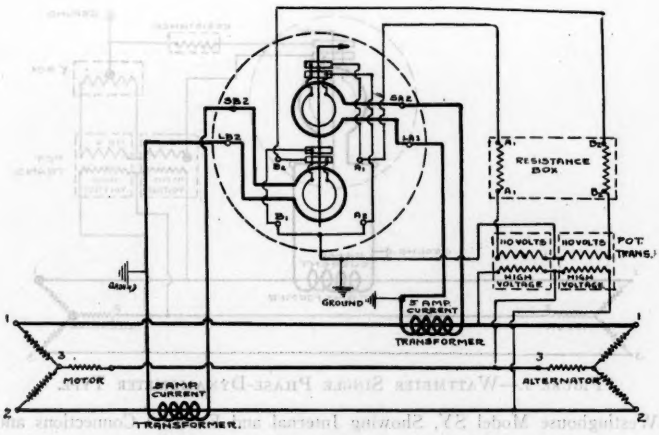


FIGURE 8.—WATTMETER THREE PHASE-DYNAMOMETER TYPE.

Westinghouse Type SY, Weston Model 368. Internal and External Connections of Meter and Resistance Box Showing Hook-up to Three Phase Power Lines Through Current and Potential Transformers.

THEORY: The operation depends on the mutual attraction and repulsion between adjacent circuits carrying current. Currents flowing in the same direction in parallel wires attract, and in opposite directions repel. As the instrument has no iron core the action is called electrodynamic instead of electromagnetic. The meter is practically two complete single phase wattmeters with the moving coils of each locked together on one shaft with a common pointer. As the total power of a three phase circuit, with either balanced or unbalanced loads, is the algebraic sum of the reading of two single phase wattmeters, in two legs of the circuit, the meter which has the two wattmeter moving elements on the same shaft automatically gives the algebraic sum of the reading. On the meter, the load currents are shown on the heavy line circuits and are the fixed field coils. The voltage coils shown in light lines are the movable coils on a common shaft to which is attached the pointer. Torsion springs are used to control the deflection of the movable coils. The force of attraction of the coils is proportional to the product of the current in the fixed coils due to the load and the current in the movable voltage coils. The inductance and eddy current losses in the meter are reduced to a neglible amount so that variation in frequency and power factor do not affect the calibration. The wattmeter automatically gives true power indications which is represented by the expression $V. \times A. \times P.F.$ The torque of the meter is the result of the instantaneous average values of the current and voltage coils and the phase displacement between the coils due to power factor of the load. Any phase displacement of the resultant field fluxes decreases the torque and thus automatically introduces the power equivalent.

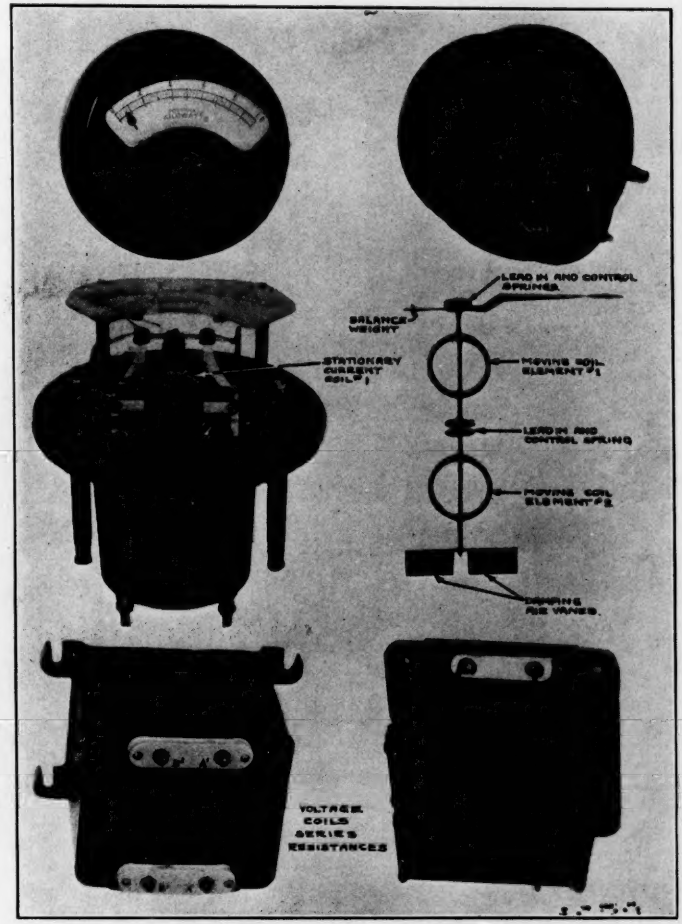

LEAD IN AND CONTROL SPRINGS
BALANCE WEIGHT
STATIONARY CURRENT COIL N° 1
MOVING COIL ELEMENT N° 1
LEAD IN AND CONTROL SPRING
MOVING COIL ELEMENT N° 2
DAMPING AIR VANES.
VOLTAGE COILS SERIES RESISTANCES

FIGURE 8a.

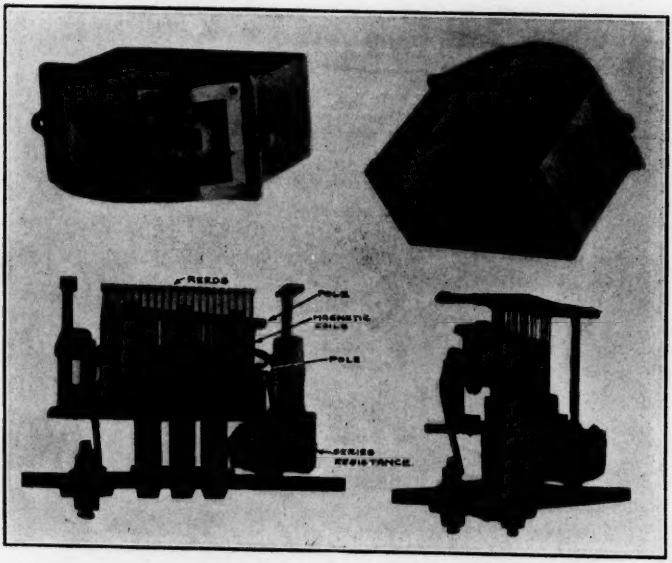


FIGURE 9a.

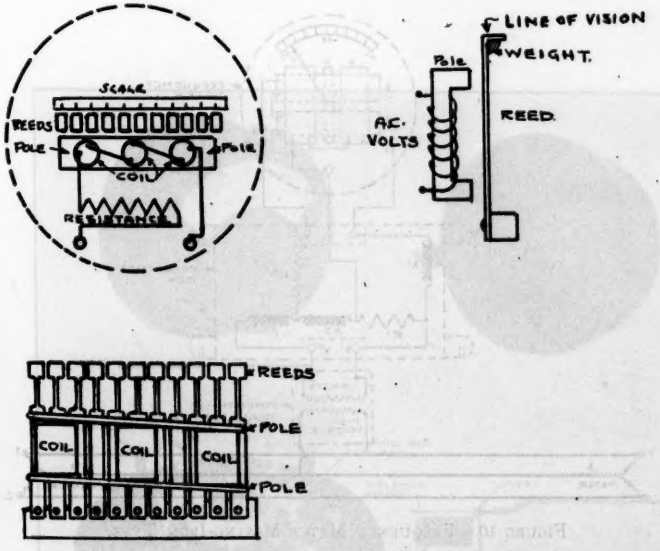


FIGURE 9.—FREQUENCY METER RESONANCE OR REED TYPE.

Hartman & Braun, German Model, Showing Operating Coils, Reeds and Internal Connections.

THEORY: Steel strips of different lengths are fastened at one end and free at the other end which is enameled white for easy vision. These strips have different periods of free vibration in the same manner as a tuning fork and can be readily caused to vibrate by outside impulses whose frequency is the same. The impulses are magnetic and are supplied by alternating current in the electromagnet which is connected to the current whose frequency is to be determined. If the free period of vibration of the reed is equal to half the period of the current, the magnetic impulses will set the reed into vibration as follows: As the current in the electromagnet increases, the reed is attracted toward the magnet and springs away as the current falls to zero; as the current increases in the opposite direction the reed is again attracted. The amplitude is thus increased with each alternation of current until the energy dissipated in the reed by molecular and air friction just equals that imparted to it by the electromagnet. If the period of the alternating quantity differs but slightly from this critical value, the impulses due to the electromagnet will not occur at favorable moments. They will occur sometimes too early and sometimes too late, so that instead of reinforcing the motion of the reed some of the impulses will oppose the motion and thus reduce the amplitude. The tuning of the reeds is done experimentally by weights on the free end of the reeds. The electromagnet is designed to be in shunt across the line and the effect of increased voltage serves only to increase the amplitude of vibration of the reed which is in synchronism with the alternations.

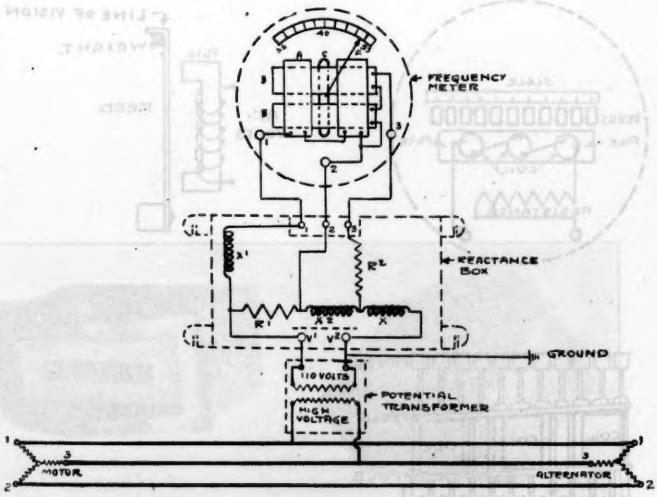


FIGURE 10.—FREQUENCY METER MOVING-IRON TYPE.

Weston Model 214, 355, Internal and External Connections of Meter and Reactance Box, Showing Hook-up to One, Two or Three Phase Power Lines Through Transformer.

THEORY: On the meter, A-A and B-B are double fixed coils mounted 90 degrees apart. C-C is the movable element consisting of a simple, soft iron needle mounted on a shaft with no control of any kind. One coil A-A in series with an inductance X_1 is connected across the non-inductive resistance R_1 and other coil B-B in series with a non-inductive resistance R_2 is connected across the second inductance X_2. The soft iron needle takes up the position of the resultant field produced by the two coils. When the frequency increases, the current decreases in A-A and increases in B-B, thus shifting the direction of the resultant field and the position of C-C to which the pointer is attached. The opposite effect takes place when the frequency is decreased. The series inductance X serves merely to dampen the higher harmonics.

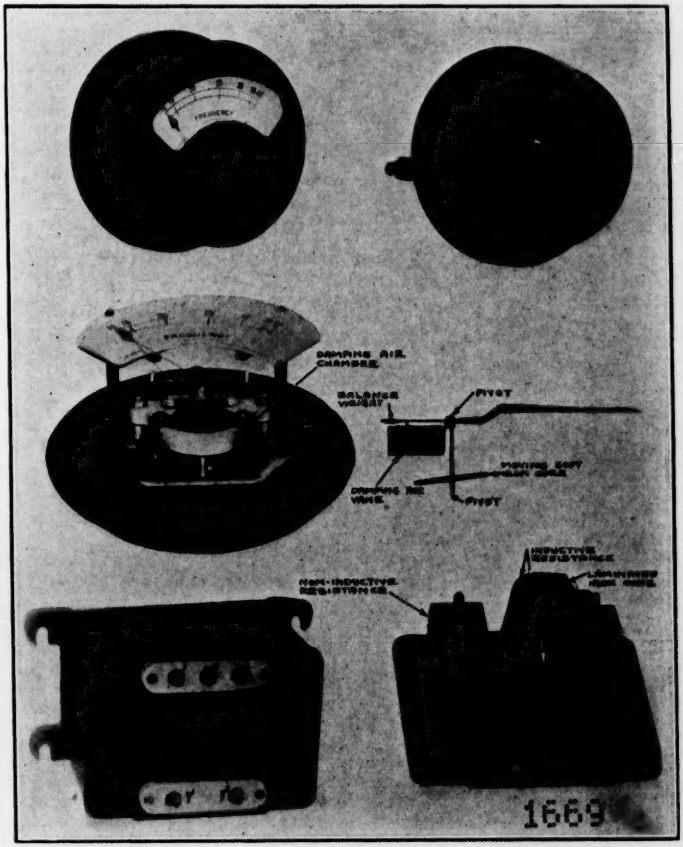

DAMPING AIR CHAMBER
BALANCE WEIGHT
PIVOT
MOVING COIL MAIN ARM
DAMPING VANE
PIVOT
NON-INDUCTIVE RESISTANCE
1669

FIGURE 108.

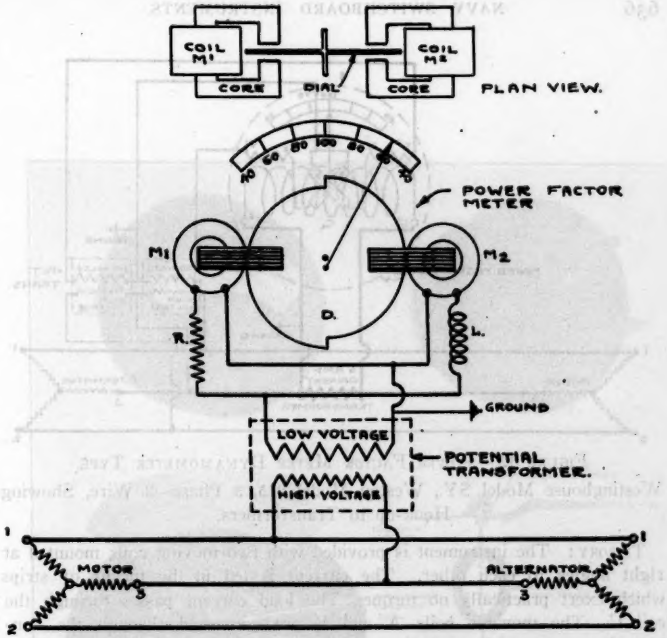


FIGURE 11.—FREQUENCY METER INDUCTION TYPE.

Westinghouse Model—5D, Internal and External Connections of Meter Showing Hook-up to One, Two or Three Phase Power Lines Through Potential Transformer.

The essential feature of this instrument is, in general, a combination of two induction voltmeters, the electromagnets of which tend to cause the disc to rotate in opposite directions. The coil M_1 of one of the voltmeter elements is connected in series with a non-inductive resistance R, and the coil M_2 of the other voltmeter element is connected in series with a relatively high inductance L. The current through M_1 is thus independent of frequency, while that through M_2 will vary inversely as the frequency, other conditions remaining constant. The coils are so adjusted that any change in voltage causes the torque due to the two coils to vary in the same ratio. The inductions of the instrument are thus independent of voltage variations. The aluminum disc D is not a true circle as shown, as any change in frequency would produce continuous rotation. The left-hand edge of the disc which moves under M_1 is practically the arc of a circle whose center coincides with the shaft. The right hand, which moves under M_2, is practically the arc of a circle whose center is slightly above the shaft. With this arrangement the amount of metal in the air gap of electromagnet M_1 is practically constant, while the amount of metal in the air gap of M_2 varies with the position of the disc. When the frequency decreases, M_2 becomes stronger than M_1 and the disc turns counter clockwise. The part of the disc in the air gap of M_2 decreases until the torques balance, when the disc stops. The opposite effects take place on increase of frequency. The exact shape of the disc is obtained by experiment and avoids the use of controlling springs.

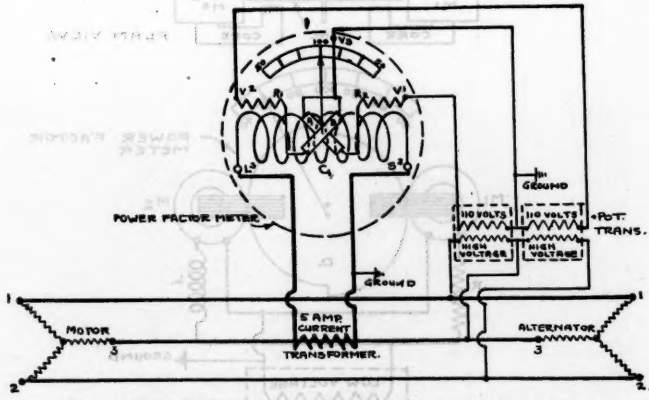


FIGURE 12.—POWER FACTOR METER DYNAMOMETER TYPE. Westinghouse Model SY., Weston Model 215, 3 Phase—3 Wire, Showing Hook-up to Transformers.

THEORY: The instrument is provided with two moving coils mounted at right angles to each other. The current is led in by means of strips which exert practically no torque. The load current passes through the coil C. The movable coils A and B are connected through the non-inductive resistors R_1 and R_2 across the line voltages as shown. They are so connected, by employing the natural phase differences in the three phase circuit, that one coil carries a current at a phase angle 30 degrees in advance and the other coil at a phase angle lagging by 30 degrees from the load current at unity power factor. Under these conditions, the two movable coils are equally acted upon by the load current in coil C by reason of symmetry, and place themselves so that they are at equal angles, or 45 degrees, with the fixed coil C. At this position the pointer is central and indicates 100 per cent power factor. At 50 per cent power factor, which corresponds to a phase angle of 60 degrees, the current in one of the movable coils is in quadrature with the load current and exerts no torque, whereas the current in the other movable coil has a component in phase with the load current and tends to move so that it is parallel to the load current coil C. This causes an indication of 50 per cent power factor at either end of the scale depending upon whether it is a leading or lagging phase angle. For intermediate phase angles, the two coils exert corresponding torques indicating the power factor. Since any variation of load current or voltage acts equally on both movable coils, the deflection of the movable coils is independent of the magnitude of the main current, or of voltage variations unless these are reduced to a very low value. As the inductance in the circuits is negligible, the meter is practically independent of frequency.

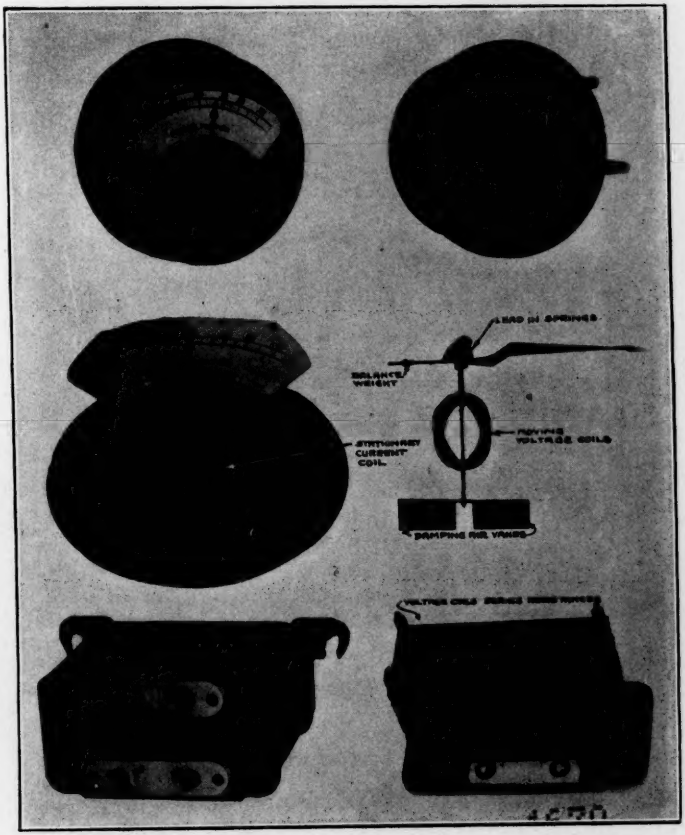

LEAD IN SPRINGS
BALANCE WEIGHT
STATIONARY CURRENT COIL
MOVING VOLTAGE COILS
DAMPING AIR VANES

FIGURE 12a.

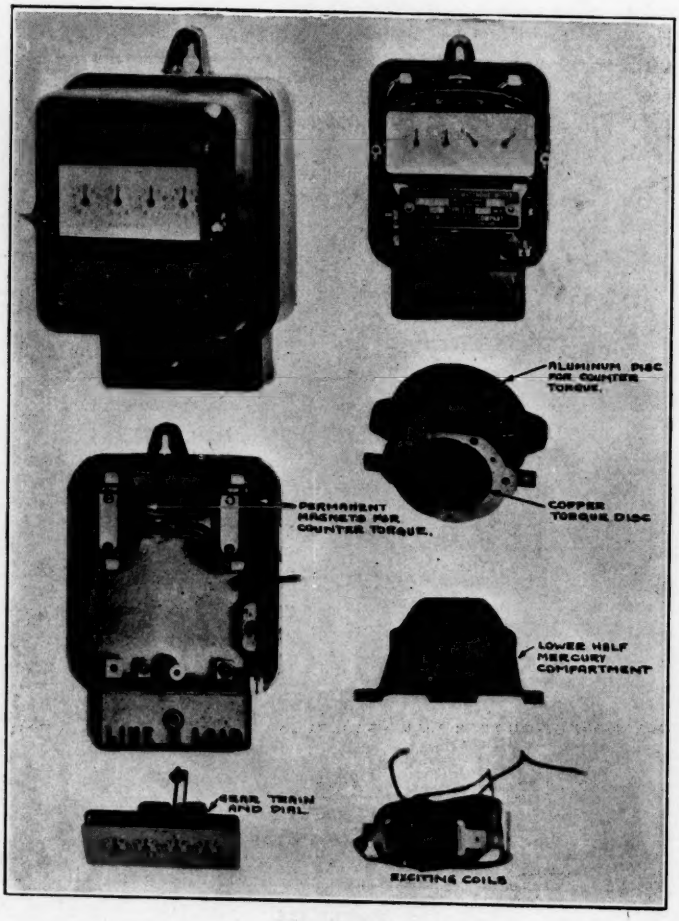

ALUMINUM DISC FOR COUNTER TORQUE.
PERMANENT MAGNETS FOR COUNTER TORQUE.
COPPER TORQUE DISC
LOWER HALF MERCURY COMPARTMENT
GEAR TRAIN AND DIAL
EXCITING COILS

FIGURE 13a.

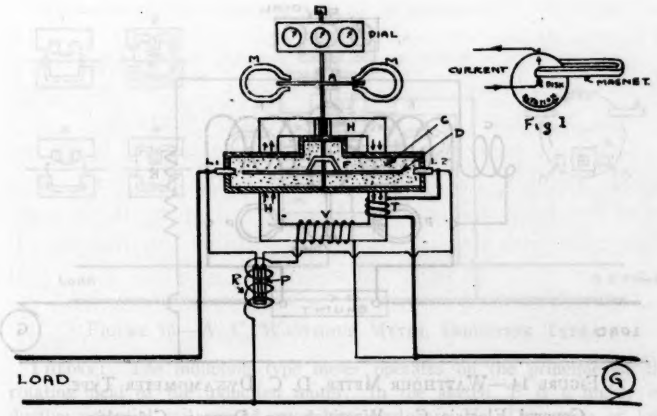


FIGURE 13.—D. C. WATTHOUR METER, MERCURY TYPE.

General Electric Co., Sangamo.

THEORY: The principle of operation of the mercury motor type of meter is best illustrated in Fig. 1. This consists of a copper disc with a shaft, free to rotate between the opposite poles of a permanent magnet. When current is passed through the shaft into the disc and out at the circumference, the reaction between the current and the field of the magnet develops a torque which causes the disc to rotate as indicated, so as to tend to carry the current out of the field. The above principle as applied to the meter is shown in the meter sketch. C is a compartment of insulating material containing a quantity of mercury in which is placed the copper disc D. The weight of the disc, shaft and auxiliary disc A is reduced to a slight upward thrust by the buoyant effect of float F. The usual lower jewel bearing is thus eliminated and the weight factor of the moving element is practically negligible. The load current passes from L_1 to L_2 through the mercury and diametrically through the disc by the shortest path. The disc being cut by flux from core H produced by shunt coil V, a torque is produced which is proportional to the current and the line voltage. The friction factor of the meter due to mercury friction at high speeds is compensated by the series turns T on the core H. The friction of the meter at low speeds is compensated by means of the added current supplied by a thermo couple P. This couple is heated at one end by the coil R in series with the shunt coil V. The cold end of the couple is joined in the current circuit at L_1 and L_2. The counter torque of the meter which is necessary to keep the disc from increasing speed under a fixed load torque is obtained by means of an aluminum disc A mounted on the shaft and revolving between two permanent magnets. The counter torque of the disc builds up due to eddy currents on the disc until at a certain speed it just balances the load torque speed.

PERMANENT
MAGNETS FOR
COUNTER TORQUE.
COPPER
TORQUE DISC
LOWER HALF
MERCURY

THEORY: The principle of operation of the mercury motor type of meter is best illustrated in Fig. 1. This consists of a copper disc with a shaft, free to rotate between the opposite poles of a permanent magnet. When current is passed through the shaft into the disc and out at the circumference, the reaction between the current and the field of the magnet develops a torque which causes the disc to rotate as indicated, so as to tend to carry the current out of the field. The above principle as applied to the meter is shown in the meter sketch. C is a compartment of insulating material containing a quantity of mercury in which is placed the copper disc D. The weight of the disc, shaft and auxiliary disc A is

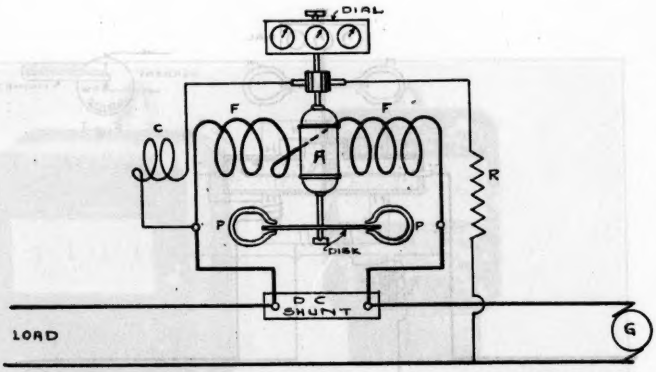


FIGURE 14.—WATTHOUR METER, D. C. DYNAMOMETER TYPE.

General Electric Co., Westinghouse, Duncan, Columbia.

THEORY: The dynamometer type consists essentially of two stationary series coils FF and a moving armature shunt coil A with sliding brushes. The rotation of the armature is caused by the action of the current flowing in armature coils which are located in the field produced by the current coils FF similar in principle to the D. C. motor, except that no iron core is used in either the armature or fields. The armature shaft on which is mounted the gear train and recording dial is free to rotate continuously. The current in the armature is proportional to the voltage of the load and the current in the field coils is equal to the load current. The torque on the armature is proportional to the product of the load voltage and current, or in other words is proportional to the power. In order that the driving torque may remain proportional to the power, there must be present a counter torque whose value increases or decreases with the load. This is obtained by means of an aluminum disc, mounted on the armature shaft, which revolves between the poles of a permanent magnet. Since the eddy currents produced on the disc due to its revolving in a magnetic field are proportional to the speed, the speed of the armature increases until it is just balanced by the counter torque of the disc. Thus neglecting friction, the speed of the armature which is indicated by the dial counter is proportional to the load. Although friction in the meter is reduced to a minimum, at light loads it would be a large percentage. This is compensated by field coil C in series with the voltage. This flux is added to that of coils FF and is adjusted to give additional torque to the armature equal to the friction factor. This type of meter can be adjusted to operate on A. C. power, but owing to inherent advantages of the induction type in its freedom from brush contacts and lighter moving element, the induction type is universally used for A. C. power while the dynamometer or commutating type described here is applied only to D. C. power.

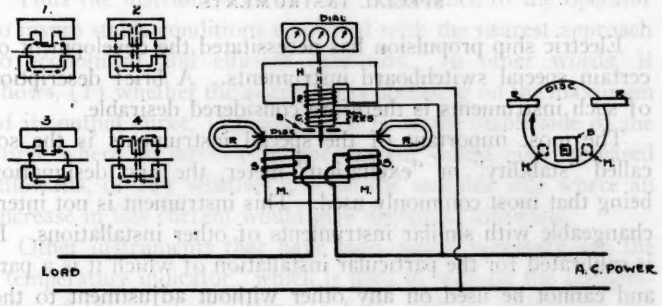


FIGURE 15.—A. C. WATTHOUR METER, INDUCTION TYPE.

THEORY: The induction type meter operates on the principle of the rotating field of the induction motor. In the sketch, P is a highly inductive potential coil wound on a laminated core H. The series or load coil SS wound on core MM is practically non-inductive. The fluxes of the series coils are in phase with the load current, while the flux from the potential coil aided by an auxiliary inductive coil C is adjusted to give 90 degrees displacement. The direction of the magnetic fields at each quarter phase of the alternating currents is shown in sketches 1, 2, 3 and 4. The resultant eddy currents set up in the copper disc of the moving system which carries the gear train to the dial mechanism, causes a rotating torque proportional to the current and the voltage of the power. The necessary retarding action or counter torque is obtained with permanent magnets RR on the same torque disc and operates to balance the speed of the disc in the same manner as described on the D. C. watthour meters. Due to the absence of any moving coils, commutator or extra retarding disc as in the D. C. meters, the weight of the moving element of the induction meter is very low and the slight friction factor is compensated for by a closed-circuit stationary copper disc B in the voltage coil air gap, the position of which is varied to give a slight reaction torque on the main revolving disc due to additional eddy currents.

SPECIAL INSTRUMENTS.

Electric ship propulsion has necessitated the development of certain special switchboard instruments. A brief description of such instruments is therefore considered desirable.

The most important of the special instruments is the so-called "stability" or "excitation" meter, the first designation being that most commonly used. This instrument is not interchangeable with similar instruments of other installations. It is calibrated for the particular installation of which it is a part and cannot be used on any other without adjustment to the new conditions. Ordinarily these instruments are interchangeable between the different units of the same installation.

In a system consisting of induction motors connected to and controlled by a variable frequency alternator there is danger of reducing the field excitation of the latter to such a low value that the system will become unstable and the motors will fall out of step. The stability meter therefore is designed to show how near the motors are to the drop-out point and enable the operator to vary the excitation accordingly.

For a given condition of operation, *viz.*, with motor speed sufficient to drive the ship at a given number of knots and the throttle of the steam end of the generator locked so that the generator will maintain a constant frequency and constant power output, a reduction in field excitation is desirable to raise the power factor and thus increase economy and improve efficiency of operation through less heating of the generator field. The temperature of the generator field is usually the limiting feature of the design.

A reduced field excitation also reduces generator voltage, and as the power output is a constant the current must increase. But a given amount of voltage is required to force a given current through an external load. The instrument thus shows that the current value for the load has the necessary amount of voltage applied.

Thus the instrument is of vital importance to the operator to insure stable conditions combined with the nearest approach to economical and efficient operation. In other words, it shows, (1) whether the alternator is operating on the maximum of its output curve, (2) whether it is on the stable side of the curve where an increase in line current would give increased kilowatts, or (3) whether it is on the unstable side where an increase in line current would give decreased kilowatts.

Other instruments that have been found necessary are the "temperature indicator" which is used to indicate the temperature of the rotating alternator field, and an instrument for obtaining the "hot spot" temperatures by various thermo-couples mounted in fixed locations in the stators of motors and alternators. The "temperature indicator" depends for operation on the change of resistance in the field windings with change in temperature. While electrical resistance is actually measured, the scale is marked with corresponding temperature indications and these represent the average temperature of the windings. The instrument is calibrated for a particular installation and cannot be interchanged with other instruments of the same type without recalibration.

With the thermo-couple actual "hot spot" temperatures are indicated by the thermal voltage generated at the fixed couples. Locations for the couples are selected with a view toward obtaining the worst conditions. They cannot be used on rotating armatures because of the interference with the accuracy if the leads must be connected through the brushes. Usually six couples located at different points are enough to locate the hottest spot of any given motor or alternator. The ordinary location is midway between the ends of the armature core, either in the bottom of the slot or between the two coils in a slot where a two-coil winding is used. Several methods are possible to obtain the temperature indicated by the thermal E. M. F., but the potentiometer is at present most used.

"Hot spot" temperatures are further obtained by a special indicator making use of an embedded search coil utilizing the idea of the "resistance" thermometer. From four to six search coils are embedded in the stators of A.C. machines—each can be plugged in on the instrument as desired. The length of the leads and the resistance of the joints is important and must be taken into account in calibrating, a consideration which limits the application of this type.

Descriptions of certain of the special instruments follow:

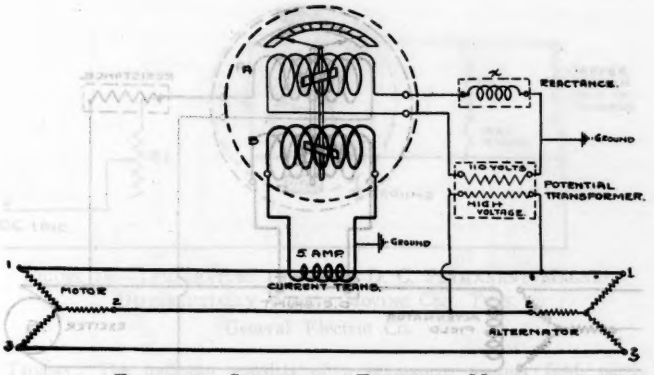


FIGURE 16.—STABILITY OR EXCITATION METER.
General Electric Co., Westinghouse E. & M. Co.

THEORY: The instrument consists essentially of a current coil B and a potential coil A, wound in opposition. The moving element consists of two soft iron vanes mounted on a common movable shaft, carrying the pointer, and uncontrolled by springs. The iron vanes pull in opposite directions under the opposed fields of coils A and B when current is on. The resultant position of the pointer then is dependent on the relative values of the flux of the opposed coils. In place of the resistance normally used with the potential coil in a voltmeter, this instrument has a reactor x of the same ohmic impedance in series with the voltage element. Thus the torque produced by the voltage element is directly proportional to the line voltage but inversely proportional to the frequency. Torque of coil B is proportional to line ampères and bucking A so that effective torque of A is also inversely proportional to line ampères. The above shows that the position of the pointer is dependent on the relation

$$\frac{\text{line volts}}{\text{line ampères} \times \text{frequency}}.$$

The above relation is a constant when the generator is working at the peak of its output curve, independent of excitation or speed. Therefore the position of the pointer is a constant under this condition, and a certain point on the arbitrary scale will always indicate such condition. Now if at maximum output and constant load and frequency the generator excitation should be, for example, increased, line volts will increase and current decrease, the voltage coil predominates and the pointer will move along that part of the scale indicating stability but lower power factor.

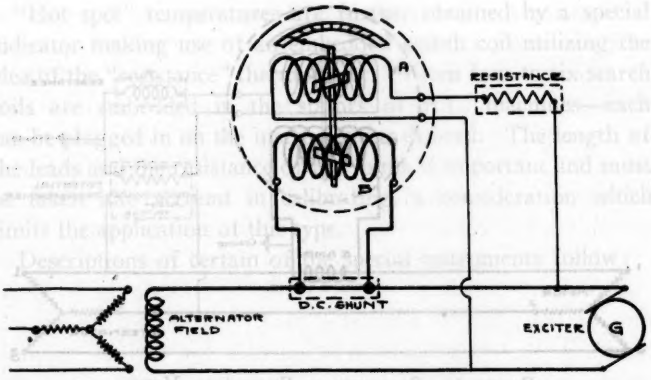


FIGURE 17.—D.C. TEMPERATURE INDICATOR, VOLT-AMPERE RESISTANCE TYPE.

General Electric Co.

THEORY: The instrument consists primarily of a moving iron type of ammeter and voltmeter mounted on the same common pointer shaft without controlling springs. The current coil B is opposed to the current in the voltage coil A. If the current to the alternator field is held constant, the voltage drop across the field will gradually build up due to increase in resistance, following temperature rise, and the increased voltage will produce more current in coil A. Voltage would, of course, be increased through hand control of exciter field.

The torque on the moving iron core of coil A will thus predominate over the torque on the core of coil B. The resultant deflection of the pointer is calibrated in temperature readings. Likewise if the voltage is held constant across the field of the alternator, the current will decrease following the increase of resistance caused by heating and the voltage coil of the meter will again predominate over the current coil and the pointer will deflect to a higher reading.

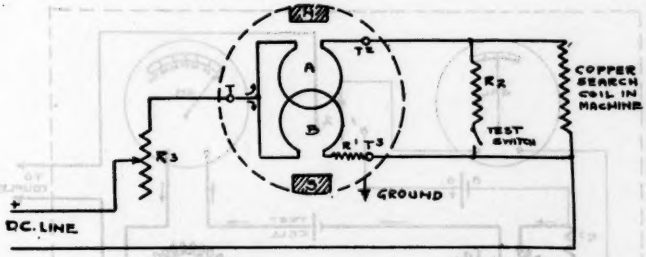


FIGURE 18.—TEMPERATURE INDICATOR, D. C. PERMANENT MAGNET DIFFERENTIALLY WOUND MOVING COIL TYPE.
General Electric Co.

THEORY: The indicator consists of a permanent magnet field, partly shown, and a moving coil consisting of a differential winding A-B on which is mounted the pointer controlled by counter torque springs, similar to a D. C. permanent magnet moving coil voltmeter except as to the differential winding and the scale marking which is usually in degrees centigrade. Current from the D. C. line passes through the moving coil and the resistance "search" coil embedded at the desired "hot spot" of the machine windings. Resistance R is adjusted so that when the resistance of the search coil is of a calibrated amount corresponding to the temperature of the lowest scale value, equal currents but in opposite directions flow around the A and B sections of the moving coil and result in zero torque, the indicator reading the lowest scale value assumed as the room temperature. Increase in temperature of the machine winding increases the resistance of the search coil which in turn causes the current in part B to predominate over that of part A. The resultant reaction of the current of coil B is thus greater than the reaction of A and the pointer moves over the temperature scale and indicates the temperature at the search coil. Resistance R, of low temperature coefficient, in conjunction with the test switch, increases the current in part A which deflects the pointer to the left for the purpose of determining whether the outfit is in operating condition and as an approximate check on the calibration. Regulating resistance R marked off in 5 or 10 volt steps is used to take care of changes in supply voltage from the line, and for accurate readings it is regulated to the nearest step corresponding to the plant voltage.

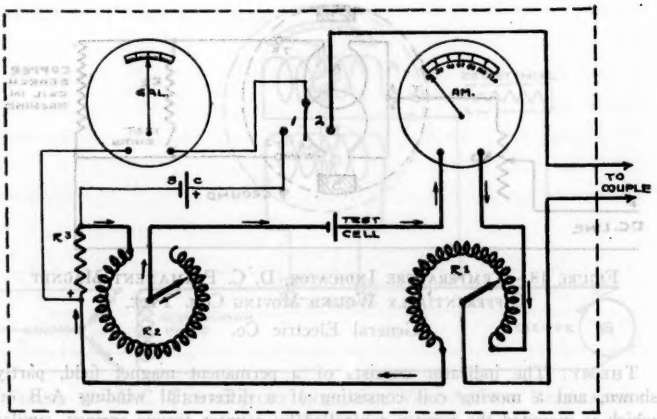


FIGURE 19.—TEMPERATURE MEASURING PANEL, POTENTIOMETER THERMO-COUPLE TYPE.

THEORY: The device consists essentially of a calibrated rheostat R_1, graduated in degrees temperature, so that when a designed current flows through this rheostat from the working test cell, the voltage drop from point to point can be varied to just balance the voltage to be produced at different temperatures by the base metals of the thermo couple employed. The setting of the desired current flows through rheostat R_1 is accomplished by two separate schemes often supplied in the same instrument, *viz.*, a sensitive low reading ammeter and the regulating rheostat R_2, or by means of a special Standard Cell SC, which has a definitely known and constant voltage. To adjust the current in R_2 by the standard cell method, consider the galvanometer circuit plugged in across connection No. 1 as shown by sketch. R_2 is regulated until drop across resistance R_3, just balances the voltage of the standard cell. R_3 is a fixed resistance designed to produce a voltage drop sufficient to accomplish this result with proper current flowing in the circuit. This balanced condition is determined by the galvanometer which has a zero center scale and its location in the circuit being such that it deflects if a difference in potential exists. When balance on the galvanometer is obtained, which means the desired current is flowing in R_1 which can be checked by the ammeter, or the ammeter relied upon alone, the test switch is thrown to position 2 and the potential from the thermo couple is brought into contact with the galvanometer on one side and the adjustable calibrated resistance R_1 on the other side. The position of the arm on R_1 is shifted until the galvanometer indicator balances again. In this condition no current flows from the thermo couple, but the voltage exactly balances the designed voltage drop on R_1 which is marked off in degrees temperature corresponding to the thermo-couple temperature. As no current flows in the thermo couple when balanced, the resistance of the leads, length of run, etc., has no effect on the accuracy of the readings. The thermo couple consists essentially of two dissimilar metals usually copper and a nickel alloy which are welded together at the "hot spot" end to form a junction. A

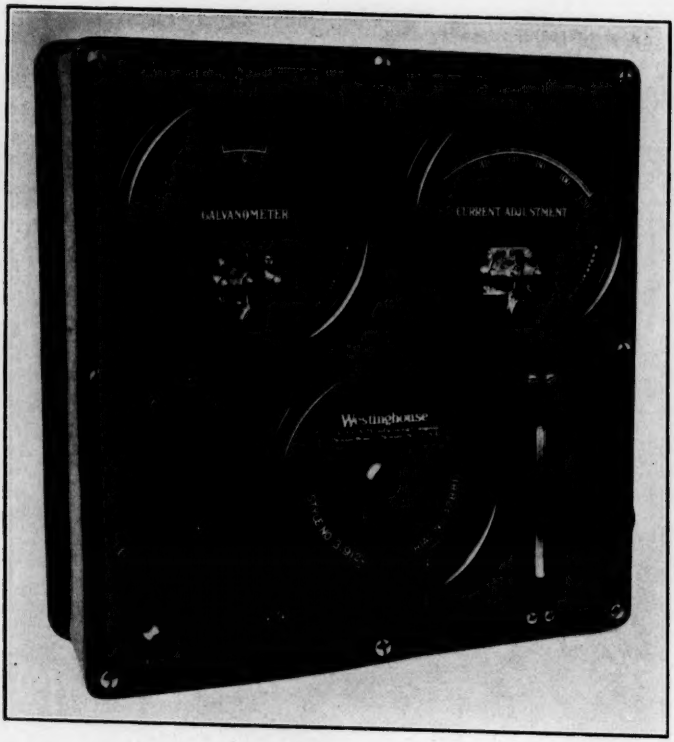


FIGURE 19a.

voltage or potential due to thermal effect will exist between the two metals proportional to the temperature, and dependent on the basic properties of the metals used in the design. The leads from the couple to the measuring device must be of the same continuous metals so that no additional thermal voltage may be introduced in the line. The instrument normally indicates the difference in temperature between the couple (hot spot end) and the meter end (cold end). In the measuring device, an auxiliary adjustment is provided to allow for varying room temperature so that the indications may give actual temperature of the "hot spot" and not the rise in temperature of the "hot spot" over the cold end of the leads in the instrument. The instrument end is assumed to be room temperature and is indicated by a mercury thermometer attached to the panel on which are mounted the complete assembly of the various units described.

PROBLEMS OF LUBRICATION RESEARCH.*

By Mayo Dyer Hersey.†

INTRODUCTION.

Some of the main physical problems of lubrication are the following:

1. The carrying power of an oil film.
2. Journal friction under varied conditions.
3. Thermal equilibrium.
4. Viscosity under pressure.
5. Oiliness

The field of lubrication offers problems for the engineer and for the chemist which are just as important as the physical problems, but which we shall not undertake to consider on this occasion. The business of a physicist is to determine the relationships between the different physical quantities governing the action of lubrication. His results may be discovered empirically or deduced theoretically, but when his work is done it can be expressed in the form of an equation and represented graphically by curves or surfaces. So far as different substances or materials are involved, the physicist is concerned only with their tangible properties (mechanical, thermal, electrical, etc.) such as can be measured in the laboratory, or represented in an algebraic formula; he is not directly interested in their constitution or origin. This on the contrary would be the business of a chemist and it is to be hoped that the chemists will succeed in telling us

* From lectures at the Massachusetts Institute of Technology, May 25, 1922; U. S. Bureau of Mines, Pittsburgh, Jan. 19, 1923, and Carnegie Institute of Technology, Jan. 26, 1923. This is not a formal research report, and such parts of it as are new will later be published in more detail.

† Recently Associate Professor of Properties of Matter, Dept. of Physics, Massachusetts Institute of Technology.

what substances will be available when some particular property is required; how such substances may be produced; and under what conditions they break down and lose their original properties. Again, it is no immediate concern of the physicist how to design or operate a bearing in the most efficient manner. He simply hands over his final equation to an engineer. The engineer, accepting the truth of the relationship established, considers how he can make the best use of it. Of course, if sufficient information is not placed at his disposal, the engineer must temporarily turn physicist himself, until this information has been discovered, and it happened to be in just this roundabout way that thermodynamics and some other essential branches of physics originated.

Now in the material here presented we shall only be concerned with establishing physical relations.

THE CARRYING POWER OF AN OIL FILM.

Osborne Reynolds showed in 1886 (Ref. 1)* that when a loaded plate A (Fig. 1) is carried along with relative velocity v over a plane surface B, flooded with oil, and kept tipped up

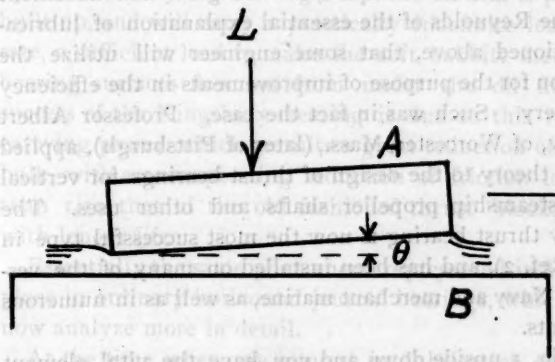


FIG. 1.—WEDGE ACTION OF OIL FILM.

*See list of numbered references at end of paper.

through some small angle θ, a wedging action is maintained which preserves a continuous film of oil between A and B. The hydrostatic pressure developed by the oil film is distributed over the under surface of plate A somewhat after the fashion of the impact pressure on the wing of an airplane. The upward resultant of this distributed pressure carries the load L. If, on the contrary, the surfaces A and B are kept parallel, the wedging action and carrying power of the film is lost, the plates stick together and there is danger of metallic abrasion.

The problem of *carrying power* consists in determining the conditions under which such a film may be maintained; and in finding out just how much load the film can support, under different circumstances as regards speed, viscosity, or other essential factors. It proves to be a more fundamental and important problem than the much more familiar one of determining friction coefficients. For a knowledge of the carrying power should enable the designer to make the total area of the rubbing surfaces a minimum, and thus to reduce the total friction.

It would be natural to expect, following the announcement by Osborne Reynolds of the essential explanation of lubrication mentioned above, that some engineer will utilize the information for the purpose of improvements in the efficiency of machinery. Such was in fact the case. Professor Albert Kingsbury, of Worcester, Mass., (later of Pittsburgh), applied Reynold's theory to the design of thrust bearings for vertical turbines, steamship propeller shafts and other uses. The Kingsbury thrust bearing is now the most successful type in service, (Ref. 2), and has been installed on many of the vessels of the Navy and merchant marine, as well as in numerous power plants.

Turn Fig. 1 upside down and you have the vital element of a Kingsbury bearing, (Fig. 2). The upper bearing surface consists of the solid disc or runner B which is now in motion with the relative velocity v. The lower bearing surface is

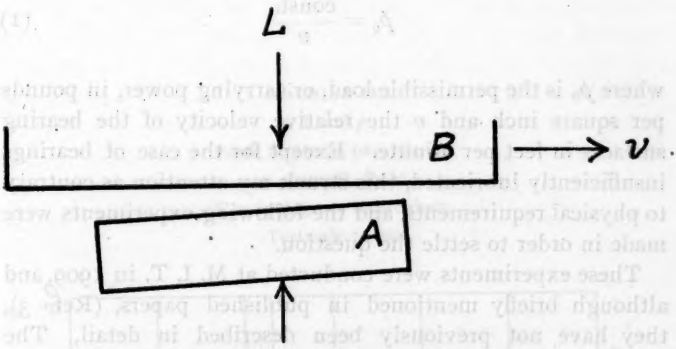


FIG. 2.—ELEMENTS OF KINGSBURY BEARING.

held stationary, and consists of an annular ring made up of a succession of pivotted blocks, any one of which may be represented by the plate A, in Fig. 1. Of course we must now understand that A is stationary, while B sweeps around toward the right. In practice, after the machine is once started in motion, the little blocks which are free to tilt take up *automatically* such an angle that the oil film is perpetually built up, and will carry a practically indefinite load; at any rate, a sufficient load to make the soft babbitt metal on the bearing surfaces flow out from under the oil film. The success of the Kingsbury bearing is due to this remarkable carrying power, which permits a great reduction of area and bulk, with a corresponding reduction of friction loss to less than 1/10 that of the old fashioned thrust block, together with longer life.

Essentially the same phenomena represented in Fig. 1 take place in ordinary horizontal journal bearings, which we shall now analyze more in detail.

At the time when my interest in lubrication experiments began, (1909), the textbooks of machine design told us that the safe load on a bearing was less at high speeds than at low Unwin, for example, laid down the rule

$$p_0 = \frac{\text{const.}}{v} \quad . \quad . \quad . \quad (1)$$

where p_0 is the permissible load, or carrying power, in pounds per square inch and v the relative velocity of the bearing surfaces in feet per minute. Except for the case of bearings insufficiently lubricated, this struck my attention as contrary to physical requirements, and the following experiments were made in order to settle the question.

These experiments were conducted at M. I. T. in 1909, and although briefly mentioned in published papers, (Ref. 3), they have not previously been described in detail, The carrying power was determined under varying conditions of load, speed, temperature, and oil-supply by measuring the electrical resistance of the oil film. Typical results are shown in Figs. 3 and 4.

Thus Fig. 3 shows the effect of gradually varying the load from nothing up to 250 pounds per square inch while running the journal (of diameter 1 inch and length 3 inches) at a constant speed of 400 R.P.M. The resistance drops from our initial value of 258×10^6 ohms, observed when the journal was caused to run in a perfectly concentric position, down to less than 1/100 as much, when the eccentricity of the journal in the bearing was the greatest that could be produced with the loads available. I do not think the break which you see in this curve has at present any great significance, the important fact being the general continuity of the curve, as revealed by the portion at the right shown on a more open scale. There was no short circuiting through the oil film; it seemed to persist under the heaviest loads.

The effect of speed in thickening up the film is shown by Fig. 4, which, again, is a typical curve selected from a series of experiments. In all cases where the oil supply was sufficient, say 60 drops per minute instead of 10, the curve never fell off, but approached the limiting resistance asymptotically as suggested by the dotted line.

MACHINE OIL.

R and p

Feed, 60 Drops per Minute.

$n = 400$ R. P. M.

T = 80 degrees Fahrenheit.

Tests 280 to 290.

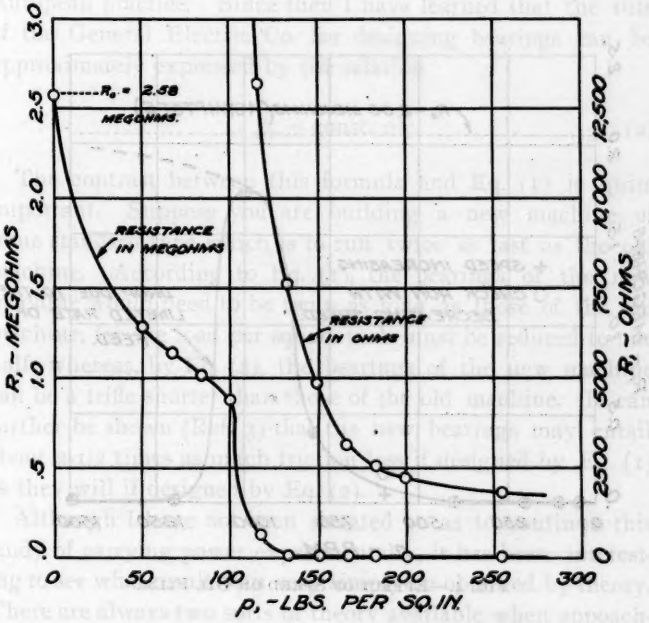


FIG. 3.—EFFECT OF LOAD ON OIL FILM.

LARD OIL.

R and *n*.

Feed 10 Drops per Minute. Temp. = 88 degrees F.

p = 200 pounds per square inch.

Tests 403–422.

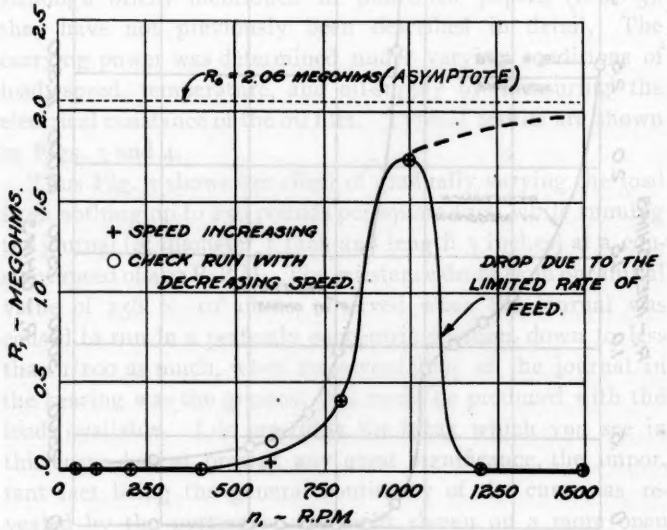


FIG. 4.—EFFECT OF SPEED ON OIL FILM.

This would seem to be convincing evidence of increased carrying power at the higher speeds in the case of bearings that are fully lubricated and properly cooled, but the results were not very favorably received at the time. It remained for Professor A. Stodola, of Zurich, to rekindle my interest in the problem four years later (1913). In the course of an encouraging interview, he assured me that the experiments ought to be continued, and that the notion of increased carrying power at high speed was in agreement with the best European practice. Since then I have learned that the rule of the General Electric Co. for designing bearings can be approximately expressed by the relation

$$p_0 = \text{const. } v^{\frac{1}{4}} \quad . \quad . \quad . \quad (2)$$

The contrast between this formula and Eq. (1) is quite important. Suppose you are building a new machine of some standard type which is to run twice as fast as the old machine. According to Eq. (1), the bearings of the new machine would need to be twice as long as those of the old machine, for the load per square inch must be reduced to one half; whereas, by Eq. (2), the bearings of the new machine can be a trifle shorter than those of the old machine. It can further be shown (Ref. 3) that the new bearings may entail about 2-1/2 times as much friction loss if designed by Eq. (1) as they will if designed by Eq. (2).

Although I have not been situated so as to continue this study of carrying power experimentally, it has been interesting to see what results can very simply be obtained by theory. There are always two sorts of theory available when appoaching a problem of this kind: Orthodox methods of integrating differential equations, which can not be carried very far without introducing simplifying assumptions that are not realized in practical bearings; and, on the other hand, dimensional theory. (Ref. 4).

Let us define the carrying power p_0 as the amount of load per unit area that will squeeze the film down to some particular thickness $x = x_0$, which we can consider a safe limit, maybe one twenty-thousandth of an inch; and let us enquire how the film thickness x may be influenced by the surface speed v, the radial clearance c, the load per unit area p, and the viscosity of the oil μ. These different quantities are indicated in Fig. 5, and if no others are involved, some physical relation like the following must exist, *viz.*

$$x = \text{funct. } (v, c, p, \mu) \quad . \quad . \quad . \quad (3)$$

The algebraic form of this function may be different for bearings of different design, but for a series of bearings which are geometrically similar in every particular, including the ratio of clearance to diameter, the ratio of length to diameter, the shape of the oil grooves, etc., it stands as a complete relation.

In such cases the principle of dimensions tells us that the equation can only be true when it is so arranged that the dimensions are alike on both sides and by applying this criterion it can be shown that Eq. (3) goes over into the form

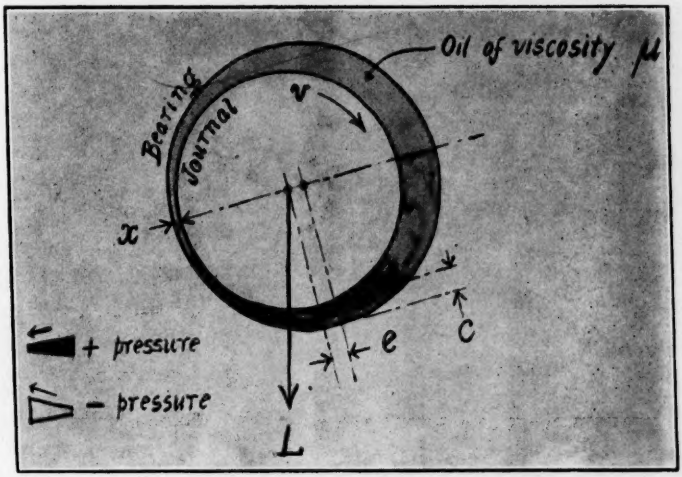
$$\frac{x}{c} = \text{funct.} \left(\frac{\mu v}{pc}\right) \quad . \quad . \quad . \quad (4)$$

Transposing so as to solve for the load per unit area we have

$$p = \frac{\mu v}{c} \text{funct.} \left(\frac{x}{c}\right) \quad . \quad . \quad . \quad (5)$$

This is a general equation true for all film thicknesses, but in the particular case where $x = x_0 =$ constant, we have by definition $p = p_0$ so for bearings of any fixed design (5) reduces to

$$p_0 = \text{const.} \left(\frac{\mu v}{c}\right) = K \frac{\mu v}{c} \quad . \quad . \quad (6)$$



x = film thickness at point of nearest approach.
c = radial clearance, e = eccentricity.
L = load = $p \times$ length $l \times$ diameter D.

FIG. 5.—CROSS SECTION THROUGH OIL FILM.

The constant in this formula depends only on the geometrical shape of the bearing, and on the safe relative film thickness x_0 / c; it is independent of the absolute size of the bearing, the speed of the journal or the viscosity of the oil.

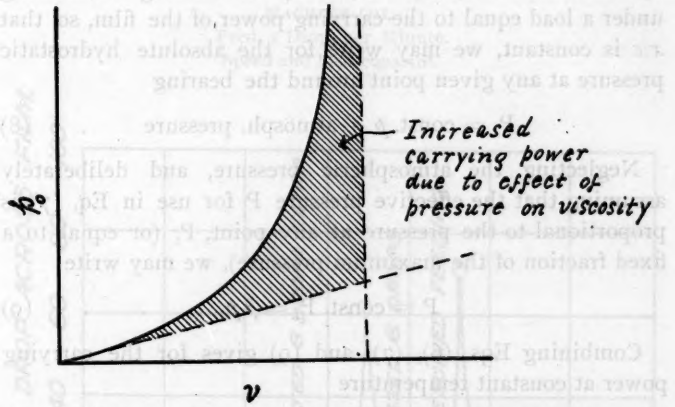


FIG. 6.—CARRYING POWER AT CONSTANT TEMPERATURE.

When you remember that the viscosity of the oil falls off at higher speeds due to heating, provided the pressure is constant, it is clear that Eq. (6), if written as a function of the speed alone, would have a very satisfactory resemblance to the General Electric formula, Eq. (2).

Viscosity, however, depends on pressure as well as on temperature, so it will be interesting to see what can be deduced from Eq. (6) if the temperature is held constant, while the effect of changes of viscosity, due to hydrostatic pressure in the film, is explicitly worked out.

Tests which will be mentioned later show that

$$\mu = \mu_0 (1 + a P) \quad . \quad . \quad . \quad . \quad (7)$$

is a reasonable approximation for moderate pressures (say up to 5,000 pounds per square inch): μ_0 being the viscosity when the absolute pressure P, equals zero; and a a constant.

Now the eccentricity of the journal in a bearing builds up an excess local pressure in the film which, at any given point, is proportional to p, the load per unit area on the bearing, but otherwise constant if x/c is constant. This can be proved by dimensional theory. If, then, the bearing is running under a load equal to the carrying power of the film, so that x/c is constant, we may write for the absolute hydrostatic pressure at any given point around the bearing

$$P_1 = \text{const.}\, p + \text{atmosph. pressure} \quad . \quad (8)$$

Neglecting the atmospheric pressure, and deliberately assuming that the effective pressure P for use in Eq. (7) is proportional to the pressure at any point, P_1 (or equal to a fixed fraction of the maximum pressure), we may write

$$P = \text{const.}\, P_1 = \beta p \quad . \quad . \quad . \quad (9)$$

Combining Eqs. (6), (7), and (9) gives for the carrying power at constant temperature

$$p_0 = \frac{Av}{1 - Bv} \quad . \quad . \quad . \quad (10)$$

in which A has been written for $K\mu_0/c$ and B for $a\beta K\mu_0/c$. The carrying power therefore goes up faster than the speed, because of the oil getting more viscous under pressure. If speeds greater than $v = 1/B = c/a\beta K\mu_0$ could be produced without raising the temperature, it would be hopeless to try to keep the film thickness down to its original value by any amount of pressure; the greater the load, the thicker the film. (See Fig. 6)

The upshot of this whole discussion about carrying power may be put in two statements:

1. The problem of carrying power simply bristles with physical questions that have not been fully worked out, and it warrants serious study because the control of carrying power is a relatively bigger factor of engineering efficiency than commonly realized.

2. It is hardly necessary, or possible, to analyze lubrication phenomena from the standpoint of obscure molecular processes until the simple consequences of the change of viscosity under pressure have first been more fully cleared up.

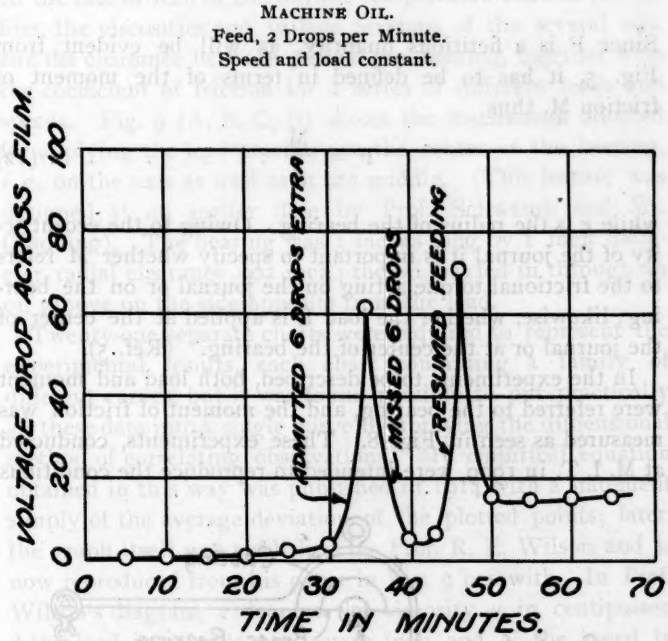


FIG. 7.—FILM THICKNESS FLUCTUATIONS OBSERVED BY ELECTRICAL METHOD.

Before leaving the subject of carrying power let me call your attention to Fig. 7, showing the remarkable delicacy of the electrical method for detecting variations of film thickness due to the admission of a few extra drops of oil.

JOURNAL FRICTION UNDER VARIED CONDITIONS.

The frictional characteristic of a bearing is commonly expressed by the coefficient of friction, a quantity which is

independent of the system of units employed, being the ratio of the tangential force of friction F to the load L, thus

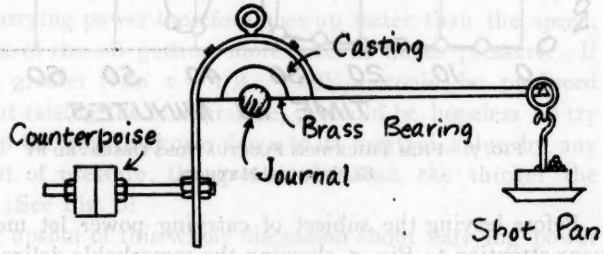
$$f = \frac{F}{L} \quad . \quad . \quad . \quad . \quad (11)$$

Since F is a fictitious quantity, as will be evident from Fig. 5, it has to be defined in terms of the moment of friction M, thus

$$F = \frac{M}{r} \quad . \quad . \quad . \quad . \quad (12)$$

while r is the radius of the bearing. Owing to the eccentricity of the journal it is important to specify whether M refers to the frictional torque acting on the journal or on the bearing; likewise, whether the load L is applied at the center of the journal or at the center of the bearing. (Ref..5).

In the experiments to be described, both load and moment were referred to the bearing, and the moment of friction was measured as seen in Fig. 8. These experiments, conducted at M. I. T. in 1909, were intended to reproduce the conditions



Friction Member. (Section K_3 of Fig. A)
(Lower half bearing added later)

FIG. 8.—MEASURING THE MOMENT OF FRICTION.

of practice more closely than usual, and yet to measure very accurately and completely the different physical factors involved. For this purpose a full bearing (rigid) instead of a half bearing was constructed and observations were recorded on the rate of feed of the oil, the temperature close to the oil film, the viscosities and surface tensions of the several oils, and the clearance between journal and bearing, together with the coefficient of friction for a series of different loads and speeds. Fig. 9 (A, B, C, D) shows the mechanism adopted for applying the load precisely at the center of the bearing, *i. e.*, on the axis as well as at the middle. (This feature was designed at an earlier date by Prof. Schwamb and Mr. Gladding). The bearing was 3 inches long by 1 inch diameter, radial clearance .002 inch; the oil was fed in through an oil groove on the side opposite from the load.

Twenty-one separate charts were required to represent the experimental results, each chart containing a family of different curves, but it was found possible to put practically all these data into a single curve by applying the dimensional method of correlating observations. My empirical equation obtained in this way was published in 1914 with a statement simply of the average deviation of the plotted points; later, the graph itself was published by Prof. R. E. Wilson and is now reproduced from his paper in Fig. 9 herewith. In Prof. Wilson's diagram, z denotes the viscosity μ in centipoises, p the load in pounds per square inch, and N the speed in R. P. M. The results should apply to any bearing which is strictly similar to mine geometrically, including eccentricity of the load if not successfully centered (An error of the order of .001 inch might account for the intercept of the graph in Fig. 9).

The dimensional theory of friction, based on the viscous drag of the lubricant, (Ref. 6), is comparatively simple, and, in the general case of forced lubrication, leads to the expression

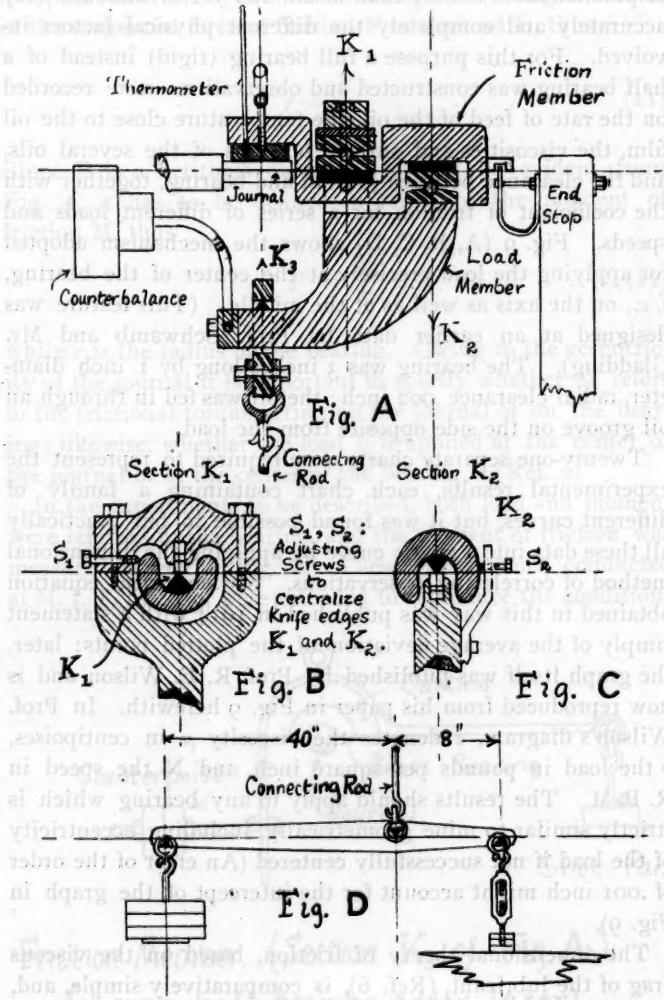


FIG. 9.—JOURNAL FRICTION APPARATUS.

$$f = \text{funct.}\left(\frac{c}{D}, \frac{l}{D}, \frac{\mu n}{p}, \frac{D^3 n}{Q}, \frac{\rho n D^2}{\mu}, \frac{p D^3}{T}\right) \quad . \quad (13)$$

for the coefficient of friction. In (13), c/D denotes the ratio of clearance to diameter; l/D the ratio of length to diameter; $\mu n/p$ may be called the generalized speed, being the product of R. P. M. by the viscosity $\mu \div$ the load per unit area, p; Q stands for the volume of oil pumped through per unit time, ρ its density, and T the external torque, if any, due to eccentric load or belt pull at right angles to axis. We can simplify this expression step by step beginning at the right and working toward the left, according to which variables we are ready to ignore (*i. e.*, to assume constant or zero). The case shown by Fig. 9 is the simplest of all, being a particular illustration of the relation

$$f = \text{funct.}\left(\frac{\mu n}{p}\right) \quad . \quad . \quad . \quad (14)$$

This assumes that there is no appreciable disturbance due to forced lubrication, and that our attention is confined to some one particular clearance ratio, c/D, and length ratio l/D, and that the load system reduces to a resultant force without any couple so that the transverse torque T of Eq. (13) is zero.

Experiments by Professor Kingsbury (Ref. 7) showed no significant departure from the recognized laws of viscous resistance down to film thicknesses of 1/40,000 inch, so I think we may regard Eq. (14) as quite rigorous for film thicknesses greater than this.

Not only is the dimensional theory easier to develop than the more detailed theories which have been published by Sommerfeld, Harrison and others, (Ref. 8), but, so far as it goes, it applies perfectly well to practical bearings with oil grooves and end losses, where the orthodox theory breaks down, (Ref. 9).

THERMAL EQUILIBRIUM.

Equations like (14) which involve the actual viscosity μ are not yet in the most convenient form for predicting the performance of a bearing, on account of the decrease in viscosity which is bound to take place after the bearing has been running for some time. It is required to deduce the corresponding equation for the performance of the bearing after thermal equilibrium has been reached.

To explain this problem, consider a very simple special case. Suppose, first, the journal is running at a high enough value of the generalized speed to be practically concentric with the bearing; in this case, it can be proved mathematically, by a simple application of the laws of viscous resistance to a layer of lubricant of uniform thickness c, that the coefficient of friction is

$$f = \pi^2 \cdot \frac{D}{c} \cdot \frac{\mu n}{p} \quad . \quad . \quad . \quad (15)$$

Suppose, second, that the viscosity at any temperature t is given by

$$\mu = \mu_0 \left(\frac{t_0 - \tau}{t - \tau} \right) \quad . \quad . \quad . \quad (16)$$

in which μ_0 was the original viscosity at room temperature t_0 while τ is a constant. Suppose, third, that the heat-transfer characteristic of the bearing is given by Newton's law of cooling

$$H = h(t - t_0) \quad . \quad . \quad . \quad (17)$$

where H denotes the heat carried off per unit time at temperature t, and h is a constant. From these three items of information it should be possible as a matter of algebra to eliminate any two quantities, and the two that we first want

to get rid of are μ and t. The remaining unknown H can now be eliminated by reference to the law of the conservation of energy, which states that, when thermal equilibrium has been reached, so that the heat is carried off just as fast as it is generated,

$$JH = Fv = fL \cdot \pi . Dn,$$

$$\therefore H = \frac{\pi}{J} D^2 l n p f \quad . \quad . \quad . \quad (18)$$

where l is the length of the bearing, and J the mechanical equivalent of heat. The final result of eliminating all 3 of the unknowns, μ t, and H, is to convert (15) into the actual working equation,

$$f = \pi^2 \frac{D}{c} \cdot \frac{\mu_0 n}{p} \left[\frac{2}{1 + \sqrt{1 + kn^2}} \right] \quad . \quad (19)$$

Clearly the factor in front of the brackets would be identical with Eq. (15) under isothermal conditions, while the bracket shows the falling off due to rise of temperature. The heating constant k is given by the expression

$$k = \frac{\pi^3}{J} \cdot \frac{\mu_0 \tau}{h (t_0 - \tau)} \cdot \frac{D^3 l}{c} \quad . \quad . \quad (20)$$

in terms of quantities already defined.

The general shape of Eq. (19) with f plotted against n is in agreement with familiar experience. But the point of our discussion is that precisely the same logic which evolved Eq. (19) can do the trick for any set of data, *i. e.*, for any type of bearing, any oil, and any cooling system. We require, in all cases, the following three items of experimental information:

(*a*) The characteristic mechanical equation for the coefficient of friction of the bearing, which contains μ, the unknown viscosity;

(*b*) The characteristic curve for the viscosity of the lubricant, as a function of the unknown temperature t; (Ref. 10);

(*c*) The characteristic formula for the cooling system, H as a function of t;

After which μ, t and H are eliminated by reference to (18). The whole matter is very much simpler than seems to be commonly supposed. That is, there is no essential mystery involved in the conditions for thermal equilibrium; certain data must be secured from experiment, but they can be obtained by three separate experiments—

(*a*) On the operation of the bearing as a mechanical system at any one film temperature.

(*b*) On the properties of the lubricant as examined at different temperatures with a viscosimeter.

(*c*) On the heat transfer characteristics of the bearing and all its surroundings as observed at different temperatures, without necessarily having the machine in motion.

The treatment just given has not yet been extended to allow for the change of viscosity under high pressure, but the general procedure would be the same.

VISCOSITY UNDER PRESSURE.

We have frequently referred to the increase of viscosity under pressure and in Fig. 11 this effect is clearly shown. Our first quantitative results, which were published in 1916, (Ref. 11) went only as far as 200 atmospheres (1 atmosphere = 1 kilogram per square centimeter = 14.2 pounds per square inch). Later the British took up this same problem at the National Physical Laboratory (Ref. 12) and went as far as 1,250 atmospheres, keeping to a constant temperature of 40 degrees C., and drew the conclusion that mineral oils showed a more pronounced effect than fixed oils when plot-

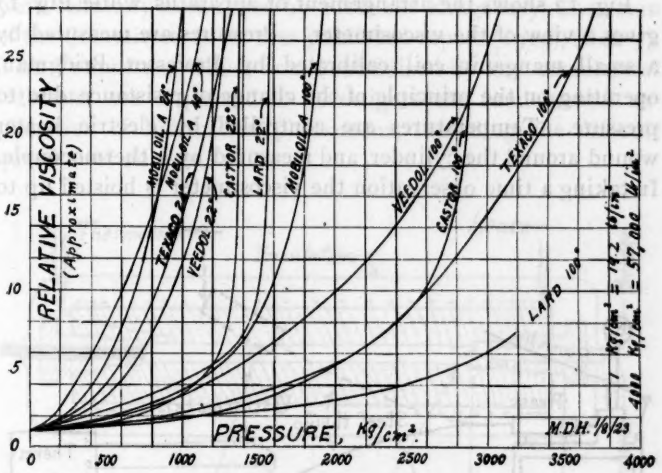


FIG. 11.—VISCOSITY UNDER PRESSURE.

ted in terms of relative viscosity. In the meantime, we had introduced the factor of temperature variation and our latest results, Fig. 11, show that at the highest temperatures and pressures (100 degrees C. and 4,000 atmospheres), fixed oils and mineral oils have a tendency to overlap. Lard oil, however, appears to retain its fluidity under higher pressure than any other thus far tried.

The viscosimeter is a modified form of Professor Flowers' rolling ball instrument, (Ref. 12), and the methods employed for producing and measuring the pressure are due to Professor Bridgman, (Ref. 14). The work was begun at Harvard, and continued at the Bureau of Standards with the assistance of H. B. Henrickson, and later at M. I. T. with the assistance of C. W. Staples and Henry Shore. The apparatus has recently been set up again in the Bureau of Mines laboratory at Pittsburgh so as to continue the observations when opportunity offers.

Fig. 12 shows the arrangement of apparatus, while Fig. 13 gives a view of the viscosimeter. Pressures are measured by a small manganin coil calibrated by Professor Bridgman, operating on the principle of the change of resistance due to pressure. Temperatures are controlled by electric heater wound around the cylinder, and measured by a thermocouple. In taking a time observation the viscosimeter is hoisted up to

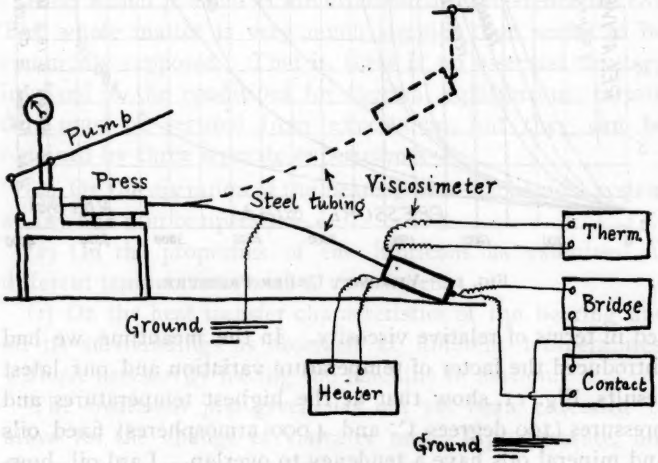


FIG. 12.—HIGH PRESSURE APPARATUS.

an angle of 15 degrees, then suddenly allowed to drop. A suitable electric contact registers the arrival of the ball at the end of its path through the oil. Absolute viscosity is very nearly proportional to the time of descent, but a more exact calibration has been made by the use of liquids of known viscosity and density.

OILINESS.

The experiments on viscosity under pressure, just described, were begun in 1915 in cooperation with the Research Sub-Committee on Lubrication, American Society of Mechanical Engineers, and they comprise one of several problems which

were definitely formulated by Professor Kingsbury, Chairman of the Committee, in its first published report (1919), (Ref. 7). This Committee, which is not subsidized by any commercial interests, was organized to investigate the fundamental explanation of lubrication phenomena, including the difficult question of oiliness.

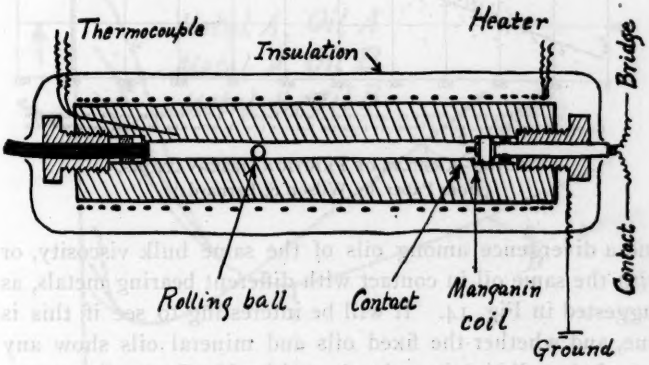
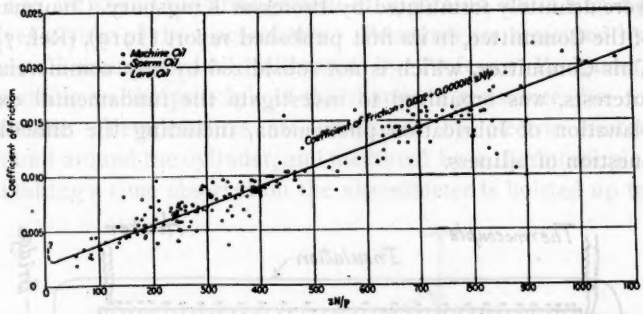


FIG. 13.—ROLLING BALL VISCOSIMETER.

The term oiliness has come into use with reference to lubrication phenomena where the film is so thin it no longer has the properties of liquid in bulk. Such phenomena occur in cutting tools and also in ordinary bearings when starting or stopping or, in general, for very low values of the variable $\mu n/p$. See, for example, Fig. 10; this curve would pass through a minimum and rise again toward the left if the necessary observations had been made. It is just in this region between the minimum point and the initial value corresponding to static friction that further investigation is most needed. Experiments at Worcester having this object in view were published by Professor Kingsbury at an early date (Ref. 15), and we understand that a new series of journal friction experiments will be undertaken by Mr. D. L. Barnard, 4th, at M. I. T. In view of Kingsbury's results, we may expect to



Results obtained by Hersey in Lubricating a Steel Journal 1 In. in Diameter and 3 In. Long and Having a Bronze Bearing with a Clearance-Diameter Ratio of 1 to 750 with Lard, Machine and Sperm Oils

FIG. 10.—RESULTS OF FRICTION OBSERVATIONS.
(from Paper by Wilson & Barnard.)

find a divergence among oils of the same bulk viscosity, or with the same oil in contact with different bearing metals, as suggested in Fig. 14. It will be interesting to see if this is true, and whether the fixed oils and mineral oils show any sort of class distinction when tested by Mr. Barnard's apparatus, and especially whether the results can be correlated with the viscosity pressure coefficients of the different oils.

As to the ultimate explanation of oiliness, a matter which requires the chemical as well as the physical viewpoint, two conceptions at least have been put forward. The first is the general idea of selective absorption, suggested some years ago by Prof. Kingsbury and supported by evidence from the action of soap and water solutions used in the lubrication of air compressors. A second idea, due to Langmuir, attributes the phenomenon to the role played by unsymmetrical molecules adhering to the metal just one layer deep. The subject has been further discussed by Bingham (Ref. 16), and by Herschel (Ref. 17). Interesting experiments by Professor R. E. Wilson, reported in Pittsburgh at the September meeting of the American Chemical Society, (Ref. 18), seem to throw the evidence strongly in favor of selective absorption of a

particular type. His papers should be read in detail by any one investigating oiliness, likewise the recent articles on Friction and on Lubrication in the Dictionary of Applied Physics, (Ref. 19).

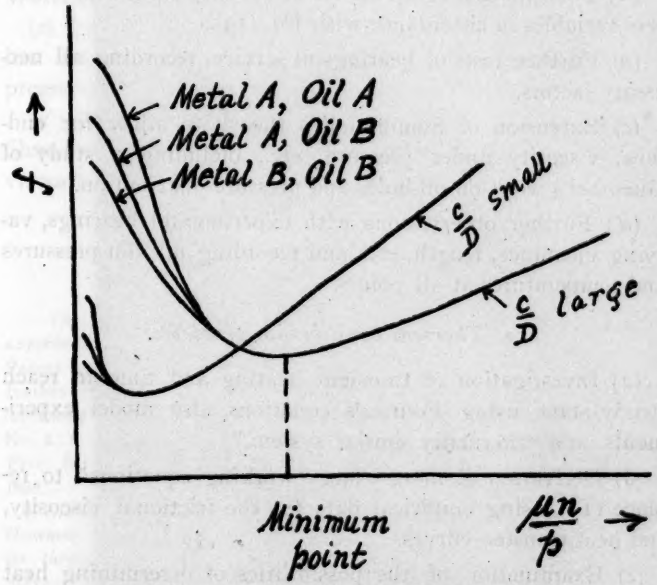


FIG. 14.—METHOD OF REPRESENTING OILINESS DATA.

UNSOLVED PROBLEMS.

The following special problems are obvious continuations of the five general problems just considered, and may briefly be tabulated without further explanation:

1. Carrying power problems.

(*a*) Investigation of other possible definitions to replace the constant-film-thickness criterion.

(*b*) Generalization of Eq. (10) to provide for the combined effect of pressure and temperature.

(*c*) Accurate film thickness measurement by optical, or electric capacity, or other methods.

2. Friction problems.

(*a*) Coordination of all available results, using dimensionless variables in accordance with Eq. (13).

(*b*) Further tests of bearings in service, recording all necessary factors.

(*c*) Extension of Sommerfeld's theory to allow for end-flow, viscosity under pressure, etc., including a study of Guembel's work on oil-holes and pressure distribution.

(*d*) Further observations with experimental bearings, varying clearance, length, etc., and recording oil film pressures and temperatures at all points.

3. Thermal equilibrium problems.

(*a*) Investigation of transient heating and time to reach steady state, using Fourier's equations, also model experiments on a "thermally similar system."

(*b*) Derivation of more exact "working equations" to replace (19), using empirical data for the frictional, viscosity, and heat transfer curves.

(*c*) Examination of the possibilities of determining heat transfer laws by simple experiments on dummy bearings satisfying the conditions for thermal similarity.

4. Viscosity problems.

(*a*) Determination of the $\mu - p - t$ surfaces for a systematic collection of samples over a much wider range of pressure and temperature, and by different methods.

(*b*) Investigation of the possible dependence of viscosity on rate of shear under extreme conditions.

(*c*) Study of plasticity at high pressure, in relation to viscosity observations.

5. Oiliness problems.

(*a*) Study of adsorption phenomena at high pressures and temperatures.

(*b*) Analysis of available results, including Kingsbury's at Worcester, from the standpoint of viscosity under pressure.

(*c*) Extension of friction experiments 2 (d) to extreme low values of $\mu n/p$, while observing the actual hydrostatic pressures.

(*d*) Development of means for measuring viscosity in thin films, and determination of minimum thickness for bulk viscosity.

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ALUMINUM AND ALUMINUM ALLOYS FOR USE ON BOARD SHIP.

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The main argument for the use of aluminum and aluminum alloys on board ship is its low weight, which is about 1/3 that of the brasses and bronzes and about 40 per cent that of iron. The reduction of weight of the apparatus on our fighting vessels has become particularly important since the international agreement on limitation of armaments, so that lighter materials will have to be employed in future design wherever possible, even at somewhat higher cost.

Aluminum at the present time is the lightest commercially fabricated metal, which may be considered for this purpose. Magnesium, pure or alloyed with other constituents, weighs only about 60 per cent of aluminum, but, aside from the fact that its manufacture has not yet reached the commercial stage, it is entirely unsuitable for use in apparatus exposed to salt atmosphere, as its resistance to corrosion is very low and no reliable method of protecting magnesium against the effect of salt atmosphere comparable to hot galvanizing, sherardizing or cadmium plating for the other metals lower down in the electrolytic potential series has so far been developed.

CASTING ALLOYS.

Pure aluminum has a very limited field of application because of its low mechanical strength and generally undesirable foundry qualities. The various possible aluminum alloys are so widely different in physical and chemical properties, that considerable care and discretion is required to select those best suitable for marine work.

Basically, aluminum alloys for use on board ship should fulfill the following requirements:

1. They must have sufficient resistance to salt water corrosion, *i. e.*, they must retain without any external protection their rated physical strength within the lifetime of a ship, so that the apparatus, in which they are used, can give the mechanical service for which it is designed as long as the ship lasts.

2. They must be chemically stable, *i. e.*, they must not disintegrate, swell, crack, or warp under any climatic or atmospheric condition, which may be met in the regular routine handling of the ship.

3. Their physical properties must correspond to those of the materials which they are to replace (brasses, bronzes, or irons), or at least come so close that the increase in section necessary to make up for the difference does not offset the possible saving in weight or bring the cost unreasonably high.

4. Their physical characteristics must be uniform and consistently obtainable.

5. They must lend themselves to established processes of fabrication and must be machinable with ordinary tools.

6. They must be "open to the trade," *i. e.*, it must be possible for any reputable manufacturing concern within the United States to use them without legal restrictions when they wish to compete for navy work.

When the matter of using aluminum on a large scale for navy work was first seriously considered by the General Electric Company, very few reliable data were found to be available regarding the suitability of existing aluminum alloys for this particular application. As the first step, all commercially known casting alloys of any promise whatever were subjected to a salt spray test, so as to exclude those which would not stand up in salt atmosphere.

This test was carried out in a "salt spray box" as standardized at the Bureau of Standards, with some slight modification. It consists of a slate box with glass cover, in which the samples

to be tested are suspended from glass rods. By use of atomizers this box is kept full of a gentle salt water mist, which acts as an extremely active corroding agent. To simulate most perfectly actual conditions on shipboard, a sea salt solution containing the same ingredients as average sea water and made up to the same strength is used. The temperature inside the box is maintained at about 35 degrees to 40 degrees C.

Eight samples of each alloy were tested. The samples were cast in the form of standard test rods, 2 to a mold, so as to be sure that each two were made of the same mixture, poured at the same temperature and in the same sand with equal ramming and gating. All samples were properly marked so as to identify mold mates, then machined all over, with threads for the testing grips and with .505-inch test section. They were then carefully weighed and measured; one rod from each mold was placed in a dry room with neutral atmosphere and its mate in the salt spray box. After 8 weeks of continuous test, all rods were pulled in a Riehle testing machine, and data on ultimate strength, proportional limit, elongation, reduction and hardness obtained. The difference in physical properties between mold mates was established and changes of weight and depth of pitting noted. We have unfortunately no positive knowledge as to how much more intense our test is than "average actual conditions." It represents a long-time continuous test under conditions about as severe as the most severe conditions met on shipboard intermittently. We generally consider that one week (168 hours) salt spray test for protective coatings on steel and iron corresponds to the lifetime of the apparatus in mechanical service, so it seemed a conservative assumption that on aluminum an 8 weeks' test should give a fair basis for evaluation of corrosion resistance in naval service. Of course, the salt spray test represents only corrosion in salt atmosphere. Coal or oil smoke with excessive sulphur content is liable to cause sulphurous acid and sulphuric acid fumes, the effect of which would not be shown by salt spray tests.

It would have enlarged the testing program beyond commercial limitations to carry out equally as extensive tests for such corrosion as for salt water corrosion, but we satisfied ourselves that in coal and oil smoke practically all aluminum alloys were at least as good as the standard brasses and bronzes.

In judging the test results, we defined the permissible limits, of course more or less arbitrarily, but guided largely by the performance of other materials which are known to be satisfactory for use on board ship as follows:

(*a*) The loss in tensile strength after 8 weeks' salt spray test must not exceed 3 per cent.

(*b*) The depth of uniform corrosion must not be greater than 2 1/2 mils.

(*c*) The unit strength of the section below the area of uniform corrosion must be the same as that of the material not subjected to the salt spray test.

The great majority of all aluminum alloys investigated had to be ruled out on their showing in the salt spray test. The binary alloys of aluminum and manganese and aluminum and silicon showed up best and came within the limits stated above.

The binary alloys of aluminum-copper, aluminum-zinc, aluminum-nickel, aluminum-tin, aluminum-magnesium, also the ternary and quarternary alloys of aluminum with copper, zinc, nickel, tin, magnesium and iron came outside the permissible limits.

In general, the corrosion is the greater the farther apart the aluminum and the alloying elements are in the electrolytic potential series. For instance, the aluminum-copper alloys showed very marked corrosion, increasing with the percentage of copper. The corrosion of the aluminum-zinc alloys was less marked.

The alloys surviving the salt spray test were then tested for chemical stability. It had been found that the best way to check up on the chemical stability of an alloy is to subject it to wet steam. Two weeks in a wet steam box will show up any tendency on the part of the alloy to disintegrate or to swell or crack.

All the alloys so tested proved satisfactory; in passing it may be said that the current prejudice against aluminum base alloys in that they are claimed to "fall to pieces" after a certain length of time, particularly when exposed to hot humid atmosphere, really hits only a small number of compositions. Aluminum-zinc alloys, with zinc (especially impure with lead and cadmium) in excess of 20 per cent, or certain aluminum-tin alloys, have sometimes shown a tendency to go to pieces, but the modern scientifically proportioned aluminum base alloys have proven perfectly stable in long service.

Since this particular prejudice seems to be especially hard to kill, a quotation from the Eleventh Report to the Alloys Research Committee of the British Institution of Mechanical Engineers (1921) may, on account of the prominence of its authors, be of value:

"In the past some doubt has been thrown on the reliability of aluminum alloys. Although this has now been largely dispelled by the wide use of these materials in war, the authors decided to examine a number of these alloys from time to time in order to ascertain what alterations, if any, took place either in the physical properties or in their dimensions. In some cases, it has been possible to test material after an interval of 10 years from the time of casting. The whole of the evidence obtained shows that within the limits of time covered by the present research any alterations in properties are accompanied by increases in both strength and hardness; in regard to shape and dimensions the alloys examined are all adequately stable for practical purposes. No signs of disintegration were seen in any of the alloys."

The next step in our investigation was to try out the casting alloys, which had so far shown up as satisfactory in actual foundry operations. The first alloy had a nominal composition of 2 per cent manganese, 98 per cent aluminum.

Its physical properties were (average): 18,000 pounds ultimate tensile, about 3,000 pounds proportional limit, 8-10 per cent elongation, 35-40 Brinnell.

We satisfied ourselves very shortly that this alloy, on account of its high shrinkage and general sluggishness, is very objectionable from the foundry standpoint. It was practically impossible, or at least very difficult, to cast any but the simplest shapes in this alloy, even with disproportionately large pouring sprues and with the most liberal use of chills and risers. Shrink-cracks and blowholes caused rejection after rejection, and we had finally to abandon its use.

We then concentrated on the aluminum silicon series which had shown up very well in the corrosion test. Silicon as an admixture to aluminum used to be considered an undesirable impurity, to be kept down and eliminated if possible. It is only a few years back that the beneficial effects of silicon on aluminum, from the corrosion and foundry standpoint, were realized and carefully studied.

Silicon was generally looked upon as a factor causing corrosion and brittleness. However, R. Frilley, about 1911, made a series of tests on silicon and its binary alloys with other commercial metals particularly to determine to what extent corrosion of aluminum is due to the silicon element. Frilley tested a number of samples for corrosion and the results are summarized as follows:

I.	Aluminum	97.42	Corrosion about equal to that of high purity aluminum.
	Iron	.26	
	Silicon	2.33	
II.	Aluminum	97.80	Corrosion 166 times as great as high purity aluminum.
	Iron	2.95	
	Silicon	.25	

These tests (checked up by our own corrosion tests) indicated that some other cause than silicon produces corrosion which Frilley attributed to iron-silicides. Silicon in itself does not detract from the corrosion resistance of high purity aluminum, thus being superior as an alloying element to copper, nickel or zinc. Its effect on elongation is less marked (*i. e.*, decrease is less) than with copper or nickel, although greater

than zinc. Its effect on ultimate tensile strength is about comparable to nickel and copper and better than zinc. One very great advantage of the aluminum-silicon alloys is due to the fact that silicon contracts on melting and expands on freezing. Solid aluminum silicon alloys are lighter than pure aluminum (at the same temperature), but the liquid aluminum silicon alloys are heavier than liquid pure aluminum. On account of this expansion in freezing, the crystallization shrinkage (percentage change in volume in changing from a liquid at the freezing point to a solid at the melting point) is materially reduced. Whereas, for instance, the crystallization shrinkage of 99.75 aluminum is 6.6 per cent and of the 8 per cent copper alloy 6.7 per cent; that in the alloy containing 7.81 per cent silicon is 5.6 per cent and in the 11.63 per cent silicon is as low as 3.8 per cent.

The pattern shrinkage of aluminum-silicon alloys is consequently less than with most other commercial alloys (approximately 1/8 inch to the foot). This, coupled with their freedom from hot-shortness, is responsible for their remarkable foundry properties. They may be cast into very thin or rapidly changing sections without cracking; they are quite free from shrinks and porosity and are leakproof and "watertight."

It has been possible to cast silicon aluminum into shapes which could not be handled in other alloys. The liability to segregation is probably lower in the silicon alloys than, for instance, in the aluminum-copper alloys, because the densities of solid and liquid during freezing are much closer together than in the case of the copper alloy.

The normal eutectic composition is usually given as about 10.5 per cent silicon, with a eutectic temperature of 576 degrees C. Junius D. Edwards of the Aluminum Company of America, however, states that recent observations have established a normal eutectic temperature of approximately 577 degrees C. and a eutectic composition of 11.6 per cent.

The silicon-aluminum alloys up to about 8 per cent can be handled in the foundry in the ordinary way; they can be melted

together either from commercially pure aluminum and silicon or a silicon-aluminum hardener may be employed.

Above 8 per cent silicon the structure of the metal begins to get coarse and crystalline with low tensile strength and elongation, so that it loses its usefulness as a structural material.

Dr. A. Pacz of Cleveland discovered, in 1920, a process to modify aluminum-silicon alloys, particularly useful for silicon contents in excess of 8 per cent, whereby the size of the silicon particles in the structure is markedly reduced and the silicon is much more finely dispersed. The eutectic is made to contain a greater amount of silicon and the freezing point of the eutectic is lower than in the "unmodified or normal" alloys prepared by melting together without the Pacz process.

The tensile strength and the elongation are improved as the dispersion of the silicon becomes finer. Some work to "modify" and thereby to improve aluminum-silicon alloys by direct electrolytic reduction or by the action of potassium fluosilicate had previously been done by Frilley, Vigoroux, and Arrivant (see Zay Jeffries, Aluminum-Silicon Alloys, Chemical and Met. Engineering, Vol. 26, No. 16, pp. 750-754), but Dr. Pacz's invention may fairly be considered the first commercially applicable process of modification.

Pacz employs a modifying agent containing about 1/3 sodium chloride and 2/3 sodium fluoride in the form of powder, which is mixed in with the aluminum-silicon melt at high temperature (910 degrees C. being the minimum at which the required chemical reaction between the aluminum silicon and the modifying agent can take place). About 1 1/2 per cent (by weight) powder is used. After the modification process has been completed the metal is allowed to cool to the normal casting temperature and poured in the molds.

Excellent physical properties are obtained from metal so prepared, superior in strength as well as in ductility to those of the unmodified alloys up to 8 per cent silicon without decreasing the corrosion resistance. The modified alloys have generally a normal composition of from 11 to 14 per cent silicon.

NORMAL ALLOY MODIFIED BY PACZ METHOD.

Fig. 1.—Normal aluminum silicon alloy containing 12.58 per cent silicon. Eutectic plus excess silicon × 250.

Fig. 2.—Same as shown in Fig. 1 modified by Pacz process. Eutectic plus excess aluminum × 250.

Due to the fact that modified alloys revert to the normal structure with its unsatisfactory physical properties if permitted to stand in the molten state for any considerable time, care should be taken that the pouring is done within 12—15 minutes after the termination of the modifying process.

The effect of the modification is shown in two micro-photographs taken from Dr. Jeffries' article referred to above. Both photographs show the structure of the aluminum-silicon alloy containing 12.58 per cent silicon, the first one made by the direct molting together of aluminum silicon and cast without any further treatment, and the second after modification by the Pacz method. In the first case, the alloy contains excess silicon plus eutectic; in the second case, it contains excess aluminum and the silicon in the finely dispersed state.

The arch enemy of the aluminum silicon alloys is iron, as iron will combine with the silicon to iron-silicides, which tend to weaken and embrittle the metal; it will also reduce the corrosion resistance. The modification process does not materially refine the iron-silicide plates.

The higher the silicon contents in the alloy, the lower the iron should be kept, so that in order to obtain the best properties the iron in the casting of 5 per cent silicon 95 per cent aluminum should not exceed 1 per cent and in a 13 per cent silicon 87 per cent aluminum it should not exceed 0.6 per cent.

This necessitates some care in the foundry to prevent the melt from taking iron from crucible, ladle, skimmer, stirrer, etc., and also the use of high grade constituent materials, aluminum and silicon, with as low an iron content as possible. Silicon can now be obtained commercially at a reasonable cost, which is guaranteed by the manufacturer to have no more than 1 per cent iron. High grade (grade A) aluminum has from 0.3 to 0.5 per cent iron, so that the maintenance of 0.6 per cent iron maximum in the casting is not difficult, provided the metal is melted in iron-free containers. Copper in the silicon-aluminum alloys somewhat improves the machining qualities, but impairs the elongation and the corrosion resistance, so it is

generally considered advisable to keep it below 0.8 per cent where these properties are important.

It should be noted that the effect of modification is largely destroyed by subsequent remelting. New melts should always be modified, whether they are entirely virgin metals or virgin and scrap.

Of course, remelting does not entirely wipe out the effect of modification, so that the addition of modified scrap to the virgin metal is always beneficial and raises the physical properties. Modified scrap should be used in as large quantities as the foundry can produce.

In the mixing of the melt, some foundry men like to use aluminum-silicon hardener, such as the 20 per cent silicon-aluminum mixture marketed by the Aluminum Company of America, which is fused in the electrolytic bath; where the constituent elements are mixed together to the proper proportions in the melting pot or furnace, care should be taken to employ only clean solid lumps of silicon or appreciable size (1 inch to 1 1/2 inches) as silicon dust and screenings are apt to be taken out in the skimmings, thereby impoverishing the melt in silicon.

On account of the high temperature, at which the modification process is carried out, care should be taken not to stir too violently, so as to keep from "beating air" into the molten metal.

The temperature control by pyrometer during the modification process, particularly the determination of the proper time at which the powder is to be added, is somewhat difficult; a simple and reliable check of temperature conditions can be obtained by throwing small quantities of powder on the surface of the melt from time to time; if the powder on striking the surface bursts immediately in bright yellow flame, the melt is too hot; if the powder, even when poked and stirred with a rod, does not show any yellow flame, the melt is too cold; the temperature is just right if yellow flames shoot up from the powder when it is poked.

Machining of silicon-aluminum alloys is not unduly difficult; the low silicon alloys machine somewhat better than the high silicon alloys, but with the proper tools and lubricants all silicon alloys can be readily handled and good finishes can be obtained.

It should be noted that the fluoride fluxes have a great affinity for the silica, which is found in all plumbago crucibles, so that the life of crucibles is reduced by the modification process unless special protective coatings or washes are used.

The physical properties of the various silicon-aluminum alloys are given below (referred to the standard, 1/2-inch test rod cast to size in green sand and machined to .505-inch diameter).

Nominal Composition	Ultimate Tensile Pds. per Sq. In.	Proport. Limit (Approx.)	Elongation in 2 ins.	Hardness Brinnell (500 Kg.)
	Min.		*Min.*	
5 per cent Si. 95 Al... (unmodified)	16,000	3,000	3.5 per cent	35
13 per cent Si. 87 Al. (modified)	25,000	6,000	8 per cent	45

These properties are not appreciably improved by any heat treatment; neither does aging produce any noticeable change.

The low silicon alloys have been used in this country in large quantities for several years with unfailingly good results. The modified alloys have been in commercial use in France and Germany for a year and a half and in this country for more than a year. A very enthusiastic report about the achievements with modified silicon alloys in France and Germany was recently submitted by R. De Fleury in the annual Exchange Paper submitted by the Association Technique de Fonderies De France to the 1923 American Foundrymen's Convention at Cleveland.

Germany and France, of course, are more vitally interested than the United States in high-class aluminum alloys, since

they can obtain copper only with difficulty, whereas aluminum is available in practically unlimited quantities, and they have, as a matter of economical necessity, nursed the foundry and engineering development of the modified silicon-aluminum alloys (although originally an American invention) through failures and disappointments which would have hopelessly discouraged the American foundrymen and engineers. They are now using the modified silicon alloys at the rate of about 100 tons a month each and are steadily enlarging the field of their applications.

The silicon-aluminum alloys, unmodified and modified, cover the greatest part of the applications met with on board ship, but there is still a big field left, where aluminum-silicon alloys could not be substituted for the heavier material due to their inherently low proportional limit and their limited ultimate tensile, which does not meet that of the high-class bronzes or of malleable iron.

For the sake of comparison we give the following table:

Composition	Ultimate Tensile (Min.)	Proport. Limit	Elongation in 2 ins. (Min.)	Modulus of Elasticity
Cast Iron	18,000	6,000	...	14×10^6
Malleable Iron	50,000	18,000	10	25×10^6
88—10—2 Bronze	30,000	9,000	16	12×10^6
80 Copper, 16 Zinc, 2.5 Tin, 1.5 Lead	20,000	5,000	12 per cent	12×10^6
5 per cent Si. 95 Al	16,000	3,000	3.5	10×10^6
13 per cent Si. 87 Al	25,000	6,000	8 per cent	10×10^6

It has unfortunately so far been impossible to produce a casting aluminum alloy which would combine higher physical properties than obtainable with silicon alloys, particularly ultimate tensile strength and proportional limit, with high resistance to salt-water corrosion. Dr. Walter Rosenhain of the British National Physical Laboratory has brought out just

before the close of the war the so-called "Y" alloy with a nominal composition of 4 per cent copper, 2 per cent nickel and 1 1/2 per cent magnesium, which in sand after heat treatment gives over 30,000 pounds ultimate tensile strength with 3-4 per cent elongation.

It is, however, quite difficult to make (on account of the nickel) and in our tests it did not come up to our requirements for corrosion resistance, so that it seemed advisable, if corrosion resistance had to be sacrificed, to investigate other possibilities first. After considerable experimentation our attention was centered on an alloy made in this country on a considerable scale which has high physical properties without the foundry difficulties of the "Y" alloy. Its nominal composition is 4 per cent copper, with magnesium from 0 to 0.3 per cent and the remainder aluminum. This alloy, after heat treatment and aging, has the following physical properties, in the 1/2-inch sand cast test rod.

Nominal Composition	Ultimate Tensile (Min.)	Proport. Limit (Approx.)	Elongation in 2 ins. (Min.)	Brinnell (500 Kg.)
4 per cent cu. ..... (without magnesium)	31,000	15,000	4 per cent	85
4 per cent cu. .... (with magnesium)	35,000	18,000	1½ per cent	100

Aging adds about 10 per cent to the tensile strength, 50 per cent to the proportional limit and decreases the elongation by about 25 per cent.

Open air-aging over a period of several months will gradually increase the physical properties produced by the heat treating and quenching, with most of it accomplished during the first week, but the same and even better results can be achieved by short-time artificial aging at elevated temperature. Machining requiring high accuracy should never be done before aging.

This alloy did not quite come up to our requirements for corrosion resistance, showing a loss of about 8 per cent tensile strength after 8 weeks in the salt spray with a depth of uniform corrosion of about 3 mils but at that it is considerably better than the ordinary No. 12 alloy (92 Al. 8 Cu.) with about 15 per cent loss in tensile strength or the other commercially used casting alloys (except the silicon-aluminum or manganese-aluminum alloys), so that we felt it could be used on board ship if its limitation is taken account of and some protective coating (non-metallic paint or electrolytically deposited cadmium) supplied.

Complete fusion of this alloy occurs at 645 degrees C. (1193 degrees F.) and complete solidification at 540 degrees C. (1004 degrees F.). The best pouring temperature is 650 degrees C.-750 degrees C., according to the size and shape of the casting. The effect of low pouring temperature on the physical properties of the alloy is particularly noticeable and it should always be the aim of the foundry man to pour it at the lowest possible temperature.

The unrestricted pattern shrinkage is 5/32 inch to the foot. The alloy does not cast as easily as the silicon-aluminum alloys; somewhat larger sprues and risers than for the silicon-aluminum alloys are required, more on the order of aluminum alloys with zinc, and rapid changes of section should be avoided.

The alloy machines vary nicely and holds threads exceptionally well, although on account of its greater stiffness more power is needed for machining than for the ordinary aluminum alloys.

The physical properties vary considerably with the thickness of section; for instance, a 1/4-inch section may show about 35,000 pounds per square inch ultimate tensile and 6 per cent elongation, whereas a 1-inch section from the same heat would show about 28,000 pounds and 4 per cent.

At the present state of the art, no aluminum alloy cast in sand is in use or to the best of our knowledge under development, which has markedly better physical characteristics than

those given in the preceding lines. Future advances in the art of alloying may and probably will produce a light aluminum mixture with properties close to malleable iron, but at the present time when such properties are demanded, recourse must be had to a different method of casting.

In the ordinary sand casting, the rate of cooling in the sand mold is so slow that the formation of large metal crystals is facilitated, which militates against the best physical properties of the metal. If the metal is cast in metal molds under the hydrostatic pressure of the metal in the pouring sprues, and just enough chilled that the time required for solidification is not sufficient to permit the growth of large crystals, a very considerable increase in all properties will result.

This process is commercially known as "permanent mold" or "gravity die" casting—the latter term sometimes being used to differentiate the process from the "pressure die" casting in which the molten metal is forced into artificially cooled metal dies under air pressure for the main purpose of obtaining high production and very accurate contours and dimensions, but not particularly high physical properties. Without trying to detract anything from the high commercial value of pressure die castings defined in the last paragraph, attention should be called to the fact that the structure of the metal obtained from pressure die castings is quite different from that in permanent mold castings.

Uniformly dense and homogeneous castings cannot be obtained by the "pressure die" process; on fracture, pressure die castings will be found to consist of a dense, closely grained outer stratum, which has been chilled against the metal die, and a porous inner stratum. Blowholes of varying size may be expected in the center of pressure die castings, particularly through heavy sections. These blowholes are only partially due to air trapped in the mold; the main and practically insurmountable reason lies in the fact that the surface is so violently chilled that it freezes before the inner section has a chance to solidify, and in the process of freezing it will

naturally draw from the hot metal on the inside. The gate leading into the die must for practical reasons be kept small, so that it ordinarily freezes before all the metal in the die has fully set; consequently no metal can be fed into the die after the first "shot" and there is no metal reserve to draw from for the shrinks in the casting.

In the permanent mold process the metal fills the mold gradually, by gravity only, thus allowing the air to escape from the mold cavity. The molds themselves are chilled only moderately—as brought out above—just enough to prevent the formation of large crystals in freezing, so that the solidification throughout the section is uniform, and sufficiently large risers and feeding gates are provided, so that, as the casting solidifies, additional hot metal is fed into the mold.

The effect of permanent molds on a representative aluminum alloy is shown in the illustration.

Naturally the cost of the metal dies for permanent molds is higher than for sand molds, although not nearly as high as for pressure die castings. According to the size and shape of the casting desired, the cost will run roughly between $150 and $350, but, on the other hand, the expense of a pattern can be saved and in the majority of cases the close tolerances and fine surface of permanent mold products will eliminate some machining operations, which would be necessary with ordinary sand castings.

On an average, in sand castings a tolerance of 1/32 inch in thickness should be allowed and in permanent mold castings plus or minus 10 mils. The weight tolerances on sand castings using metal pattern equipment are plus or minus 3 per cent, with wood patterns plus or minus 5 per cent and with permanent mold castings plus or minus 2 per cent.

The surface finish and general contours of permanent mold castings, while not as good as with pressure die castings, are perfectly good enough to obviate the necessity of machining, except where very close dimensions, small holes and screw threads are required.

Grain size produced by casting in iron mold (50 × dia.)

Grain size of same alloy cast in green sand (50 × dia.)

(From paper by Zay Jeffries on "Aluminum Castings" presented at Spring meeting May, 1920 of A. S. M. E. at St. Louis, Mo.)

The actual foundry cost (exclusive of molds and machining) of permanent mold castings is less than bench or machine made sand castings, unless the casting is so small that a gated pattern with more than one or two parts on one gate can be used for the sand castings. The speed of production in actual number of molds is greater with permanent molds than with sand molds, and cheaper labor can be employed, since less muscular effort and skill is required.

The present commercial limitation of size is about a 2-foot diameter circle; larger molds can be made if necessary, but cost and mechanical difficulties will be greatly increased.

In the design of castings for permanent molds particular care should be taken that in any section the ratio between metal and cooling surface does not become too small, otherwise the surface of a section with too low ratio may become solidified before the mold has had a chance to fill. Not all aluminum alloys lend themselves to the permanent mold process, as for the best results low shrinkage, high hot tensile strength and fluidity are necessary. All three alloys given in this article are suitable to the permanent mold process. However, the 13 per cent silicon alloy on account of its tendency to revert to the unmodified state, if kept in the molten condition for more than 15 minutes, which would be unavoidable with permanent mold routine, should have some copper and manganese added, making the total copper in the casting about 1 per cent and the manganese about 1/2 per cent. The alloy thus constituted can be modified in the original melt and can then be remelted and kept in the molten condition for one or two hours without deteriorating appreciably. The corrosion resistance is slightly lowered, but is still within the original requirements. In the permanent mold, as brought out above, all physical properties are improved considerably above those of sand castings from the same composition, increasing the tensile strength and proportional limit about 15 per cent and the elongation about 25 per cent. Hardness is also materially increased.

In the case of the non-reversible 13 per cent silicon alloy the properties in permanent mold are about as follows:

Ultimate Tensile	35,000
Proportional Limit	8,000
Elongation in 2 inches, per cent	5
Hardness Brinnell	65

The lowered ductility is due to the extra copper.

The fabricated aluminum alloys (forgings, sheets, etc.) will be discussed in another article to follow shortly.

THE EFFECT OF CHANGES OF DISPLACEMENT ON DESTROYER PERFORMANCE.

By Capt. P. B. Dungan, U. S. N., Member.

It has been recognized that a change in displacement in any vessel is accompanied by corresponding changes in the horsepower, speed and revolutions for any given condition. The amount of these various changes has, however, been somewhat obscure and not definitely known. While a vessel equipped with facilities for accurately measuring the above variables could approximate the extent of these changes under various conditions of loading, their accurate determination could only be attained by a series of carefully conducted trials with other conditions maintained approximately constant.

The practical effects of the change of displacement on destroyers are greater than is encountered on most other classes of vessels, as the changes in displacement encountered in ordinary service are exceptionally great as compared with the normal displacement. It is not uncommon for one of these vessels having a mean trial displacement of from twelve hundred to thirteen hundred tons, to operate under normal service conditions with an excess displacement over the above figure of from three hundred to four hundred tons; the excess displacement thus amounting to twenty-five per cent or over. Unless the effect of this increased displacement in decreased speed, decreased revolutions and increased horsepower are known; a fair and dependable comparison between the performance of different vessels cannot be made unless they are constantly operating at approximately the same displacement. This latter condition rarely obtains.

The building program for destroyers, begun during the war, was so extensive, and the need for the vessels so urgent, that it was impracticable to carry out standardization trial on all

vessels. As a result only one vessel of each type, built at the same yard, was required to be standardized. The standardization results from this trial were applied to the remainder of the vessels concerned. Only the vessels carrying out the full trials were equipped with torsion meters for the determination of horsepower developed.

Trial displacement is invariably materially less for these vessels than the displacement obtaining under ordinary service conditions. As a result the data given on the standardization curves is not directly applicable to ordinary service performance. If the displacement is materially different these curves furnish only an approximation of the actual conditions as regards speed, revolutions and horsepower.

This condition was particularly noticeable when these vessels were subjected to final trials. It was impracticable to bring the vessels to the mean trial displacement obtaining at the preliminary trial, and the excess displacement frequently amounted to twenty per cent or more, of the preliminary trial displacement.

Under these conditions most vessels were unable to equal the revolutions obtained on the preliminary trial, and the horsepower and speed taken from the standardization curves were correspondingly low and therefore unsatisfactory. When torsion meters were available the horsepower developed was frequently found under the above conditions to be greatly in excess of anything developed during the preliminary trials.

In an endeavor to determine the amount of variation due to excess displacement the Engineering members of the Board of Inspection and Survey prepared curves from the best data available from which corrections to the horsepower, speed and oil consumption could be applied to these factors as taken from the standardization curves applicable to the vessel. These curves, with full explanation in regard to their derivation and use, were published by Lieutenant Commander J. M. Irish in the JOURNAL for February, 1922.

Subsequently the Bureau of Engineering prepared rather more elaborate curves showing the performance to be expected from various types of destroyers under different conditions of displacement.

The above curves were used by the Board of Inspection and Survey in determining the performance of these vessels on final trials, and were most useful in arriving at a reasonable judgment as to their fulfillment of contract requirements.

While the above curves were believed to be approximately correct, it was most desirable to obtain accurate and definite information; and to this end a series of progressive standardization trials were authorized to be carried out with the U. S. S. *Brooks* (No. 232), built by the New York Shipbuilding Corporation, representing the Bath type of hull; and with the U. S. S. *Putnam* (No. 287), representing the Fore River type of hull. In order to insure uniformity and accuracy as far as practicable, these trials were ordered to be conducted under the supervision of the Board of Inspection and Survey.

The trials on both vessels were run on the measured mile course at Rockland, Me. Revolutions, speed and horsepower were determined in the usual manner for standardization trials. The vessels were docked in Boston and bottoms cleaned and painted just prior to leaving to begin the trials.

The trials on the *Brooks* were begun on July 8th, 1922, but were discontinued on July 9th, due to machinery troubles and the evident inaccuracy of the torsion meters. The vessel returned to Boston for the correction of these defects. After redocking the trials were again started on July 28th and continued until completion on August 3rd.

The trials on the *Putnam* were begun on August 15th, but were discontinued on the same day due to damage to the L. P. turbine blading. The vessel returned to Boston where turbine repairs were made. The trials were again started on October 5th, but were again interrupted due to failure of blades in the L.P. turbine. After repairs to the blading in both L.P. turbines, the trials were again resumed on December 3rd and car-

ried through to completion. The readings of the torsion meters were found, on working up the data, to be so inconsistent that the horsepower observations were valueless. The curves for the *Putnam* trials are therefore not included in this article.

It may be noted that curves involving displacement, speed and revolutions are consistent with those obtained on the trials of the *Brooks*.

Full standardization runs were made on the *Brooks* at each of the following displacements: (*a*) 1610.9 tons, (*b*) 1491.7 tons, (*c*) 1433.9 tons, (*d*) 1342.7 tons, (*e*) 1299.8 tons, (*f*) 1268.5 tons.

In addition to the above, partial standardization runs, consisting of runs at approximately 15 knots, 22½ knots and 30 knots, were made with the vessel trimmed down by the head. These runs were made at the following displacements: (*g*) 1602.8 tons, (*h*) 1408.5 tons.

The displacements given above are those obtaining at the middle of the high speed group of runs in each instance. This displacement is of course incorrect for the runs at the other speeds, depending upon the amount of fuel oil and other consumables expended. In order to obtain the correct displacement at the middle of the runs at other than the highest speed corrections were made to the displacement determined upon getting under way up to the time of the middle of the runs at each speed. This was done as follows: Careful draft readings were taken before getting under way and again upon anchoring, by reading the internal and external draft gauges. The readings of the internal draft gauges were then plotted on a fore and aft diagram of the vessel. A straight line was then drawn through the points representing these readings and extended to cut the perpendiculars at the position of the outside gauges. This gave the corresponding readings for the external draft gauges which was used simply as a check. The difference between the reading of the middle internal draft gauge and the draft as measured to the above line on the diagram at that point, was taken as the "hog" for the condition of load-

ing at that displacement. Corrections for the displacement as taken from the diagram were made for hog, trim and density of water. The density of water was determined at the beginning and end of each day's runs by hydrometer readings of samples of water taken from well forward of the water discharge openings in the vessels hull. Owing to the slight motion of the surface water, which always existed to a greater or less degree, the readings of the outboard gauges were not absolutely dependable, and these readings were used only as an approximate check. When conditions were favorable the readings of the outboard gauges, as checked against the readings obtained as above from the diagram, checked within an eighth of an inch or less. The draft and trim as determined in the above manner for the various trials are as shown in table A. The date of the trial and the displacement obtaining at the middle of the high speed runs are entered in this table in order to permit easy identification with the actual data curves on Plate I.

The data as observed on the two trials when the vessels were trimmed down by the head indicates that this change of trim has but little effect on the performance of the vessel.

The actual data as observed on these trials is plotted on the curves shown in Plate I and Plate IA.

The calculated displacement for the middle time of each group of runs is entered for the spot plotted for the group in each instance. These displacement corrections were obtained as follows: During the time the vessel was under way for the day her displacement decreased a certain amount. The number of horsepower hours developed during this entire time was calculated, and a factor obtained representing a flat decrease of displacement per horsepower-hour. The displacement correction was then applied to the displacement upon getting under way according to the number of horsepower-hours developed up to the time of the middle of each group of runs. This method of course is not absolutely accurate, but after careful consideration of the variables involved was decided to be the best method available, and to be sufficiently accurate for the

purpose. Fuel oil and water expenditures were taken as a check, both from the readings of meters and tank soundings.

In view of the unavoidable changes of displacement during the progress of the runs the curves in Plate I are not satisfactory for practical use.

Using the data obtained from the curves on Plate I and Plate IA, the cross curves shown on Plates II and III were drawn, the first showing displacement plotted against speed at constant revolutions; the second displacement against horsepower at constant speed.

From these cross curves the usual curves of horsepower and revolutions, and of speed and revolutions, were plotted for displacements varying from 1,200 tons to 1,700 tons. These curves are shown in Plate IV. From these curves the horsepower and speed can be read directly from the revolutions for any displacement within the limits given above.

The curves on Plate III have been drawn as a straight line. This is a slight deviation from the points actually plotted as shown. A curve drawn through the actual points would lie somewhat above the straight line for the lighter displacements and somewhat below the straight line for the heavier displacements. As this feature is fairly consistent for the curves at various speeds it is probable that the straight line is not as truly representative of conditions as the slightly wavy line. Since drawing the above curves this feature was mentioned by Admiral Dyson as having been observed in previous investigations. Except at the lowest speeds the maximum difference between the straight line curve as drawn and the curve exactly following the points is less than 1 per cent of the horsepower involved. It was on account of this slight difference that the straight line curve was drawn.

An interesting feature is indicated in the curves on Plate II where at the lower speeds an increase of displacement from the lightest conditions results in an increase of speed at constant revolutions. This same tendency to recurve is evident at the higher speeds and would probably show to the same

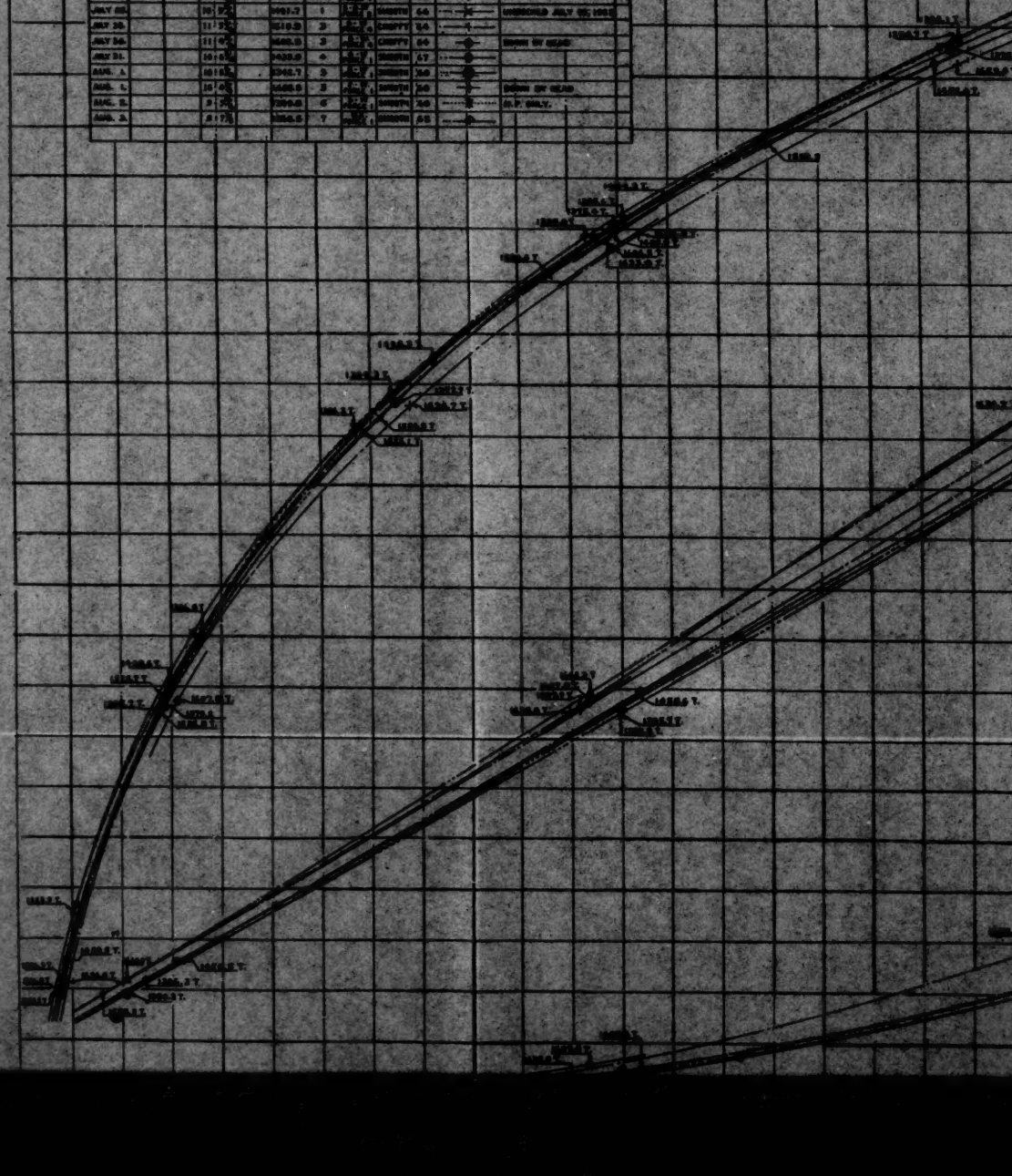
BOARD OF INSPECTION AND SURVEY
NAVY DEPARTMENT
TO ACCOMPANY REPORT OF TRIAL BOARD DATED
1935.

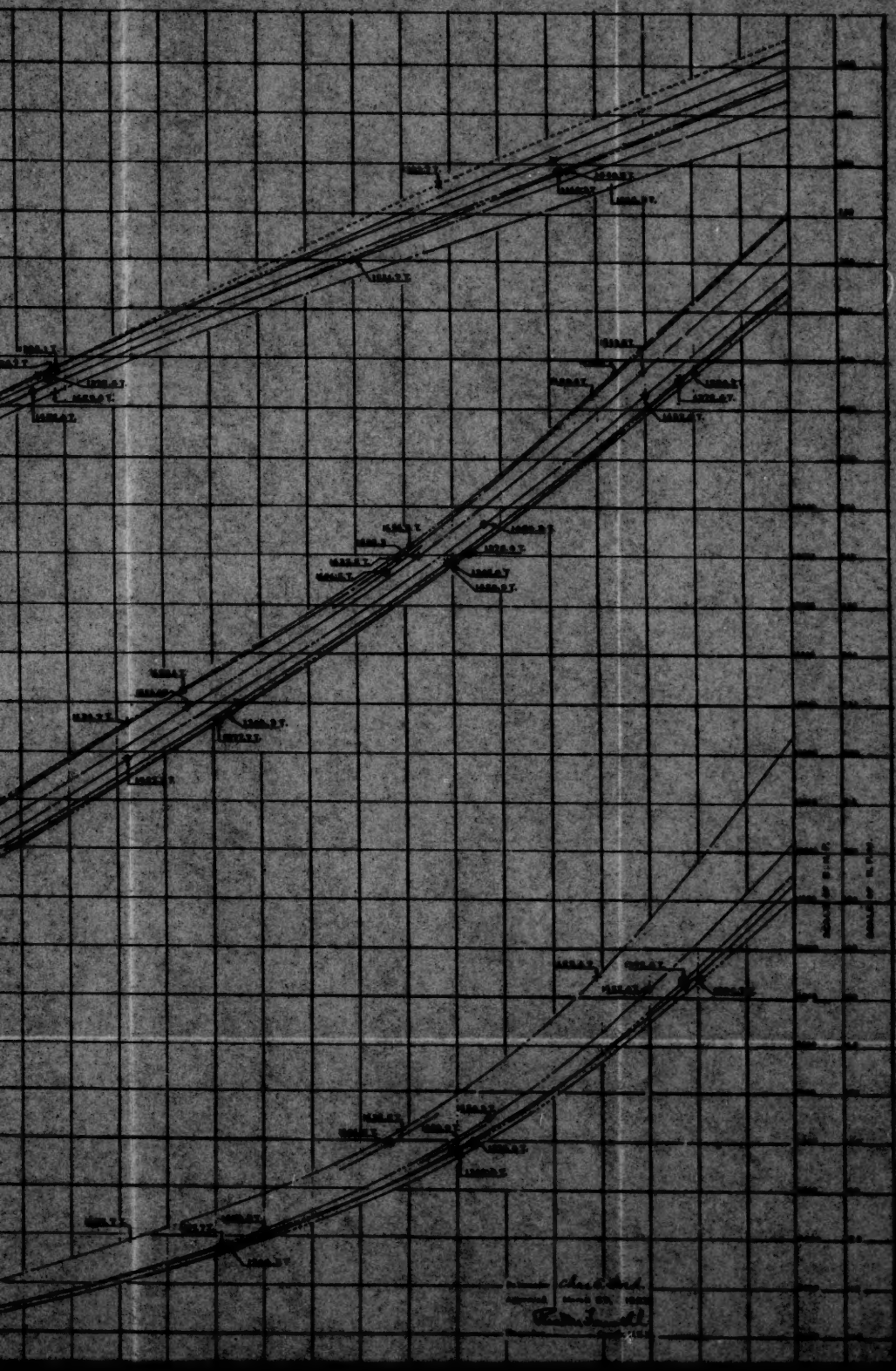
U.S.S. BROOKS.

STANDARDIZATION TRIALS AT VARIOUS DISPLACEMENTS TO DETERMINE INCREASED RESISTANCE TO PROPULSION DUE TO INCREASE IN DISPLACEMENT.

TRIALS CONDUCTED OVER THE MEASURED MILE COURSE, RICHLAND SHORE.

DATE	CONDITIONS REGARDING AMOUNT OF FUEL AND WATER ON BOARD			TRIAL	DISP.	NO. DAYS OUT OF DRY DOCK	SPEED	AREA	PROPELLER RPM	REVOLUTIONS FOR CURVES	REMARKS
	DRAFT FWD	DRAFT AFT	DRAFT MEAN								
1935 MAY 23.					1307	18		SMOOTH	60		INCREASED MAY 17, 1935
JULY 2.					1448.0	2		CHOPPY	61		
JULY 3.					1436.0	3		SMOOTH	66		
JULY 05.					1401.7	1		SMOOTH	60		DECREASED JULY 05, 1935
JULY 30.					1319.0	2		CHOPPY	56		
JULY 26.					1449.0	3		CHOPPY	64		DOWN BY HEAD
JULY 31.					1438.0	4		SMOOTH	67		
AUG. 1.					1340.7	5		SMOOTH	60		
AUG. 1.					1468.0	2		SMOOTH	60		DOWN BY HEAD
AUG. 2.					1388.0	6		SMOOTH	60		H.P. SHAFT.
AUG. 3.					1368.0	7		SMOOTH	58		





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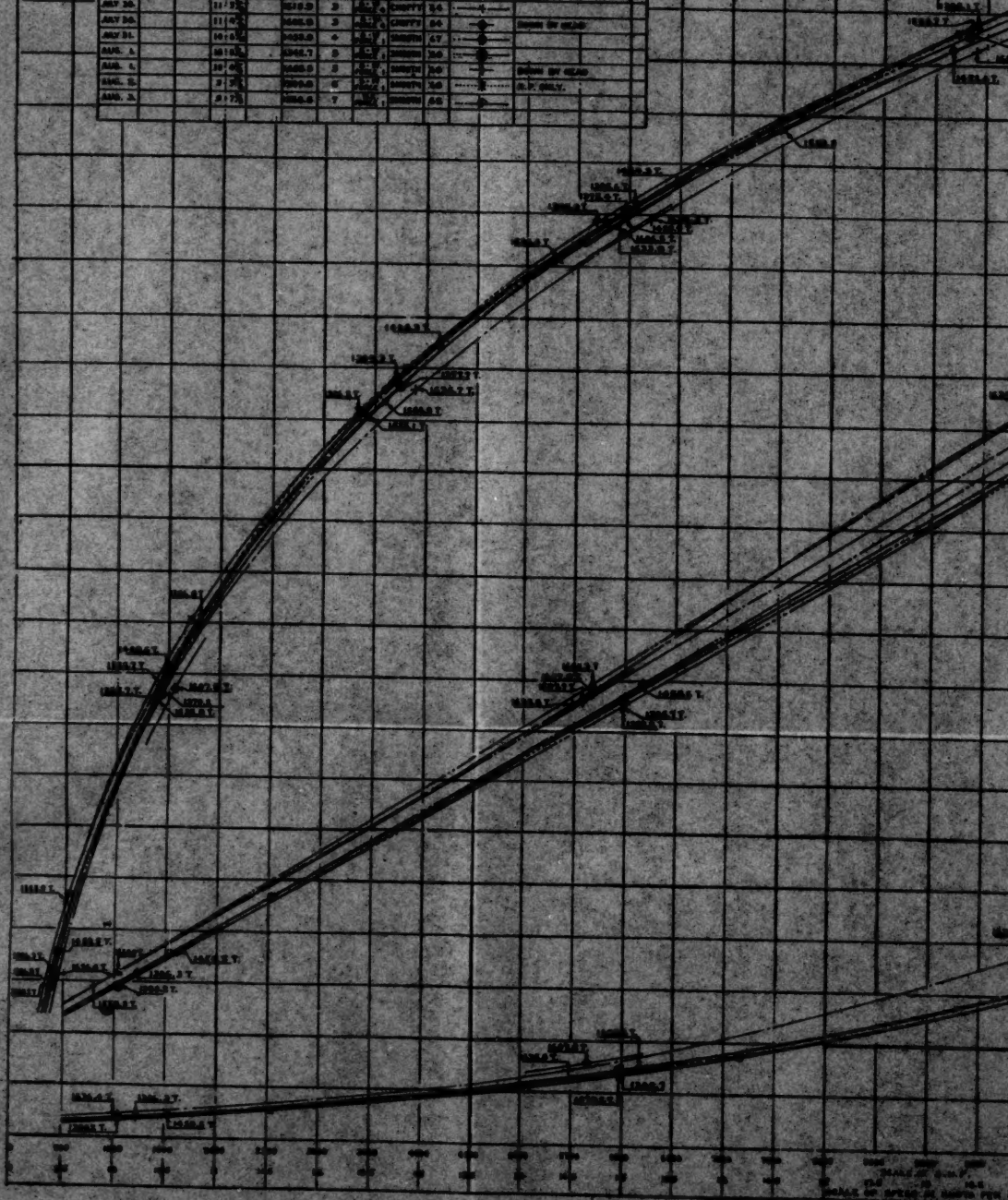
U.S.S. BROOKS.

STANDARDIZATION TRIALS AT VARIOUS DISPLACEMENTS TO DETERMINE INCREASED RESISTANCE TO PROPULSION DUE TO INCREASE IN DISPLACEMENT.

TRIALS CONDUCTED OVER THE MEASURED MILE COURSE, PACIFIC COAST.

DATE.	CONDITIONS AS TO TRIM AND DRAFT OF WATER WHEN TRIAL WAS RUN. DRAFT FORE	DRAFT AFT	DRAFT MEAN	TRIM	DISPLACEMENT IN TONS OF 2240 LBS.	SPEED	SEA	TRIALS FOR CURVE.	REMARKS
MAY 17.					12987	10	SMOOTH	68	LIGHTENED MAY 17, 1906
JULY 6.					14460.0	8	CHOPPY	67	DISPLACED JULY 6, 1906
JULY 6.					14336.0	3	SMOOTH	66	
JULY 6th.					14101.7	1	SMOOTH	66	DISPLACED JULY 6th 1906
JULY 30.					14310.0	3	SMOOTH	66	
JULY 30.					14460.0	3	CHOPPY	66	
JULY 31.					14458.0	4	SMOOTH	67	TRIM BY HEAD
AUG. 1.					14345.7	3	SMOOTH	66	
AUG. 1.					14460.0	3	SMOOTH	66	TRIM BY HEAD
AUG. 2.					14460.0	6	SMOOTH	66	S.P. ONLY.
AUG. 2.					14460.0	7	SMOOTH	66	



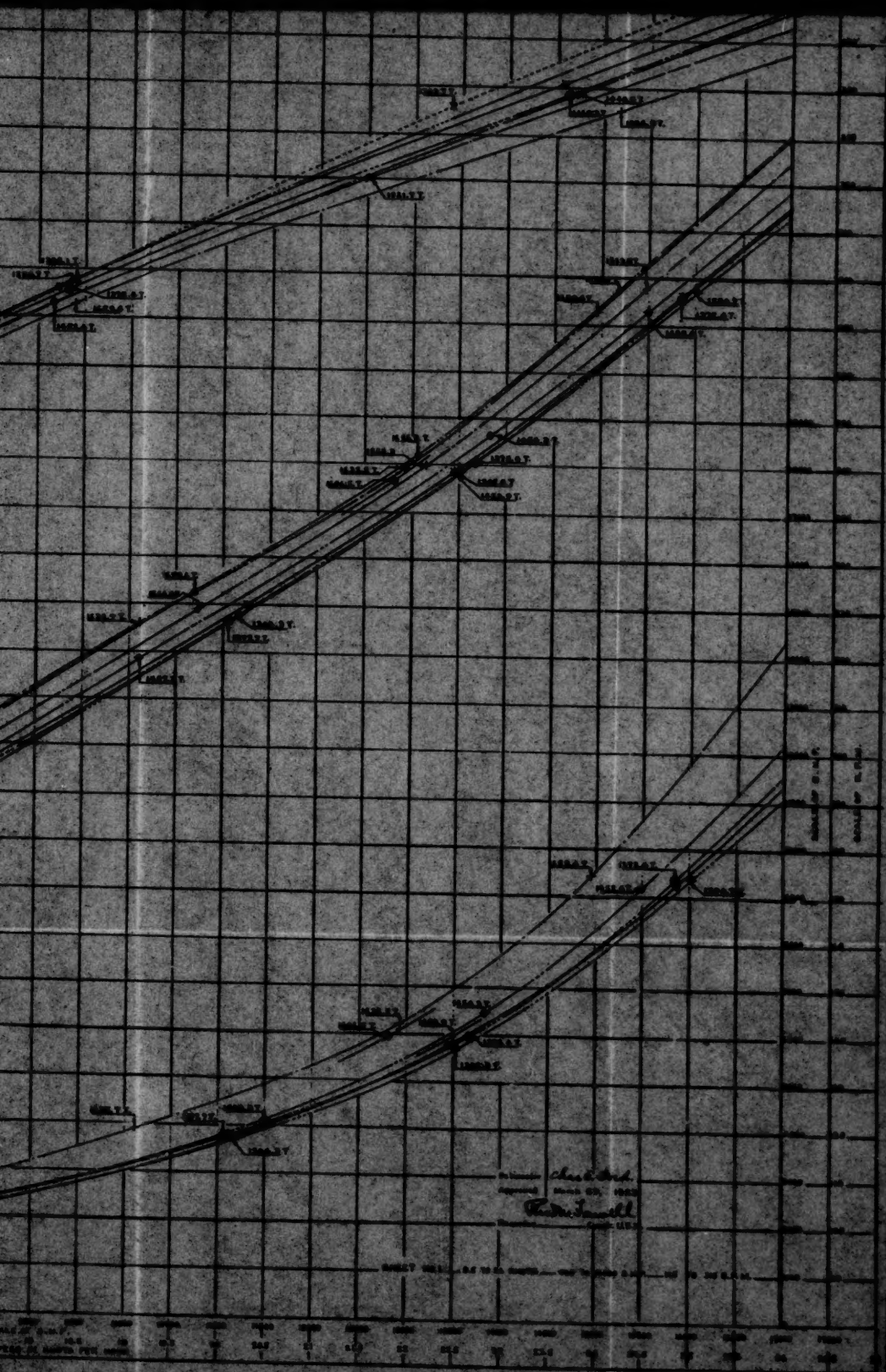


PLATE No 1

BOARD OF INSPECTION AND SURVEY

NAVY DEPARTMENT

TO ACCOMPANY REPORT OF TRIAL BOARD DATED

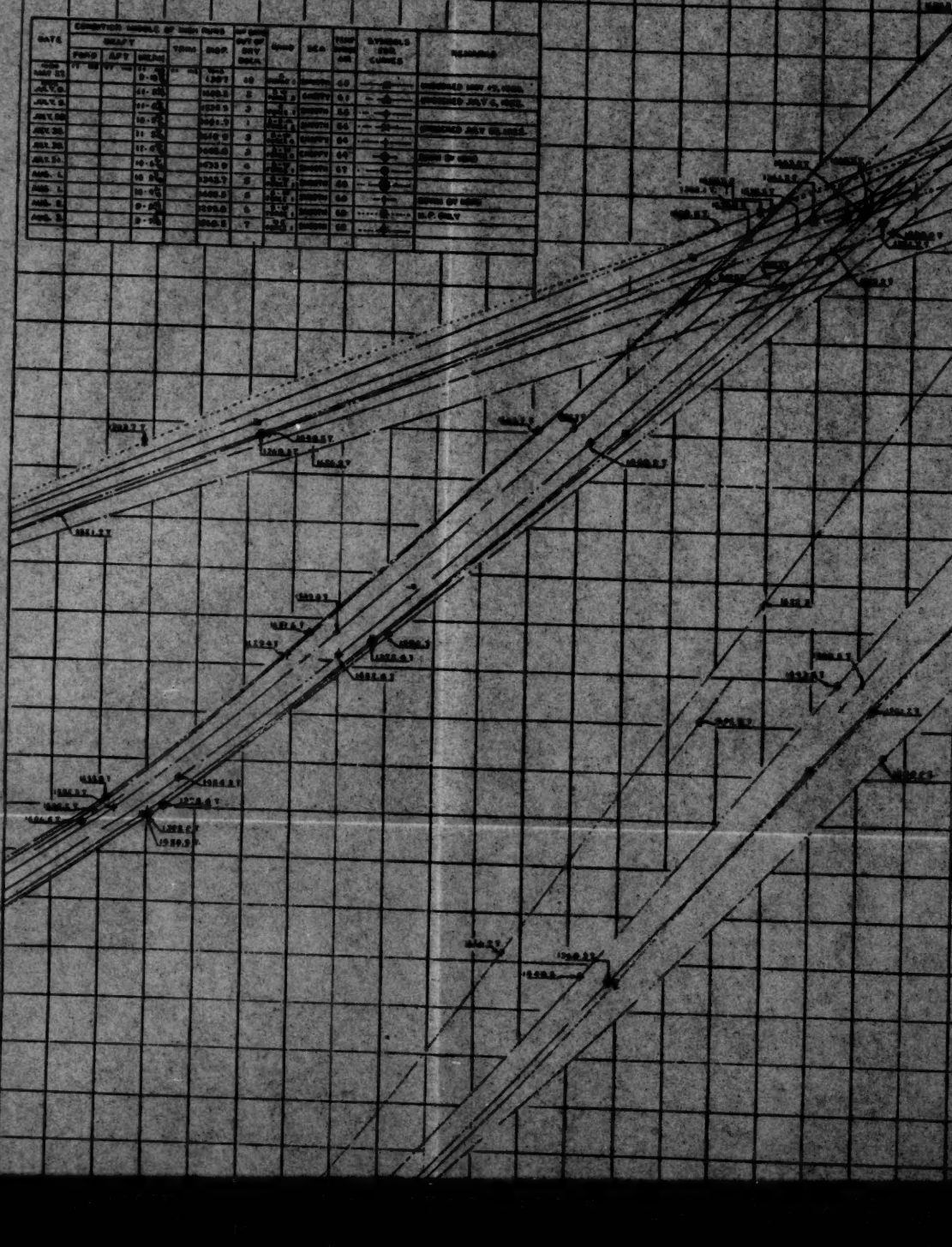
1942.

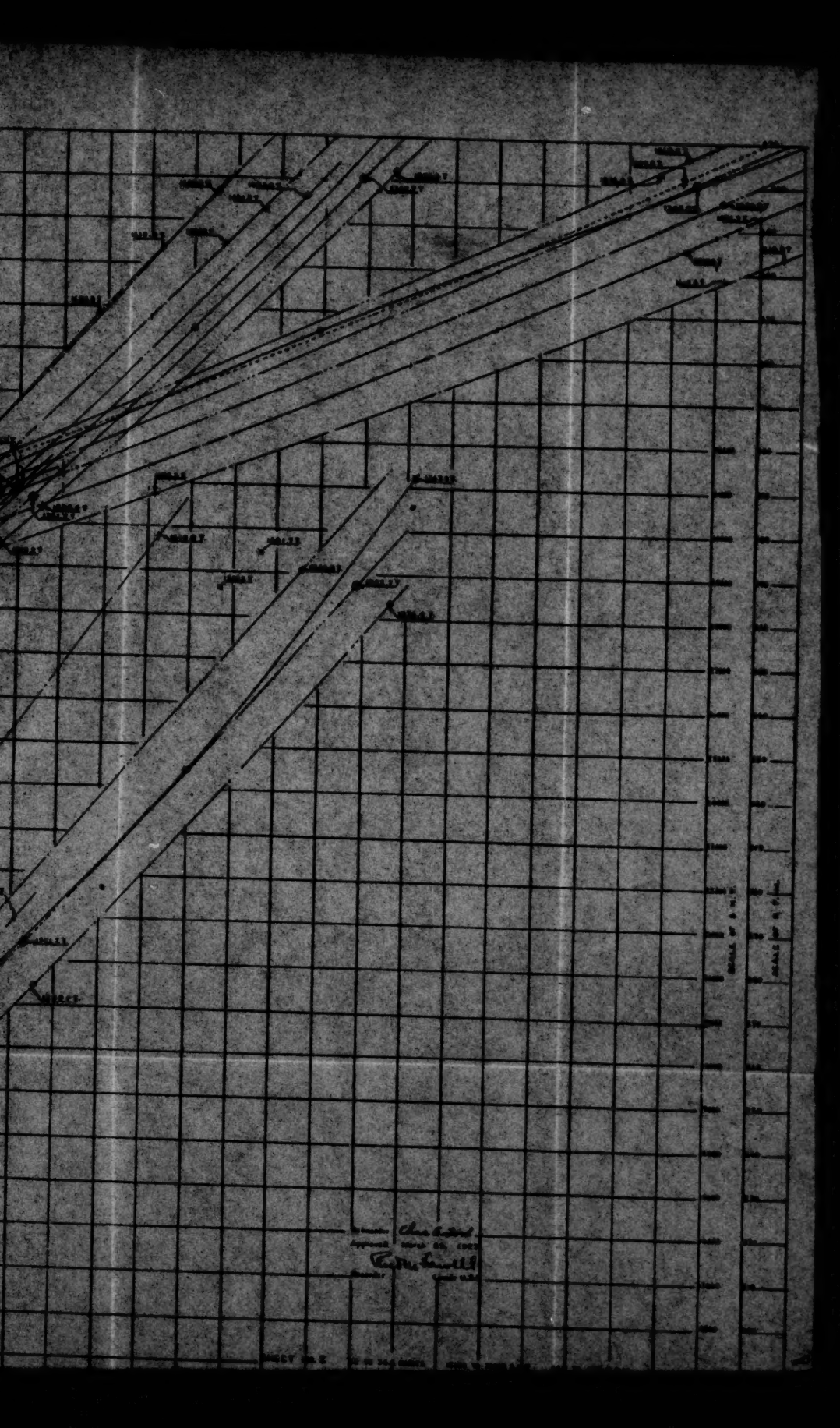
U.S.S. BROOKS.

STANDARDIZATION TRIALS AT VARIOUS DISPLACEMENTS TO DETERMINE INCREASED RESISTANCE TO PROPULSION DUE TO INCREASE IN DISPLACEMENT

TRIALS CONDUCTED OVER THE MEASURED MILE COURSE, MAINLAND SOUND

DATE	CONDITION (GENERAL OF HIGH POINT) DRAFT FORD	DRAFT AFT	DISPLACEMENT	TONS	DISP.	DAY'S RUN OUT OF DRY DOCK	SPEED	SEA	TEMP AIR	SYMBOLS FOR CURVES	REMARKS
JUNE 25			0-05		1397	40		SMOOTH	60		
JULY 6			11-05		1408.5	5		SMOOTH	61		
JULY 8			11-05		1398.5	3		SMOOTH	66		
JULY 20			10-11		1393.0	1		SMOOTH	64		
JULY 24			11-05		1410.0	3		SMOOTH	64		
JULY 30			11-05		1408.0	5		SMOOTH	69		
JULY 31			10-10		1403.0	4		SMOOTH	69		
AUG. 1			10-06		1395.7	5		SMOOTH	68		
AUG. 1			10-05		1400.0	5		SMOOTH	68		
AUG. 2			9-05		1398.0	6		SMOOTH	68		B.P. ONLY
AUG. 3			9-05		1396.5	7		SMOOTH	68		





NAVY DEPARTMENT

TO ACCOMPANY REPORT OF TRIAL BOARD DATED

1942.

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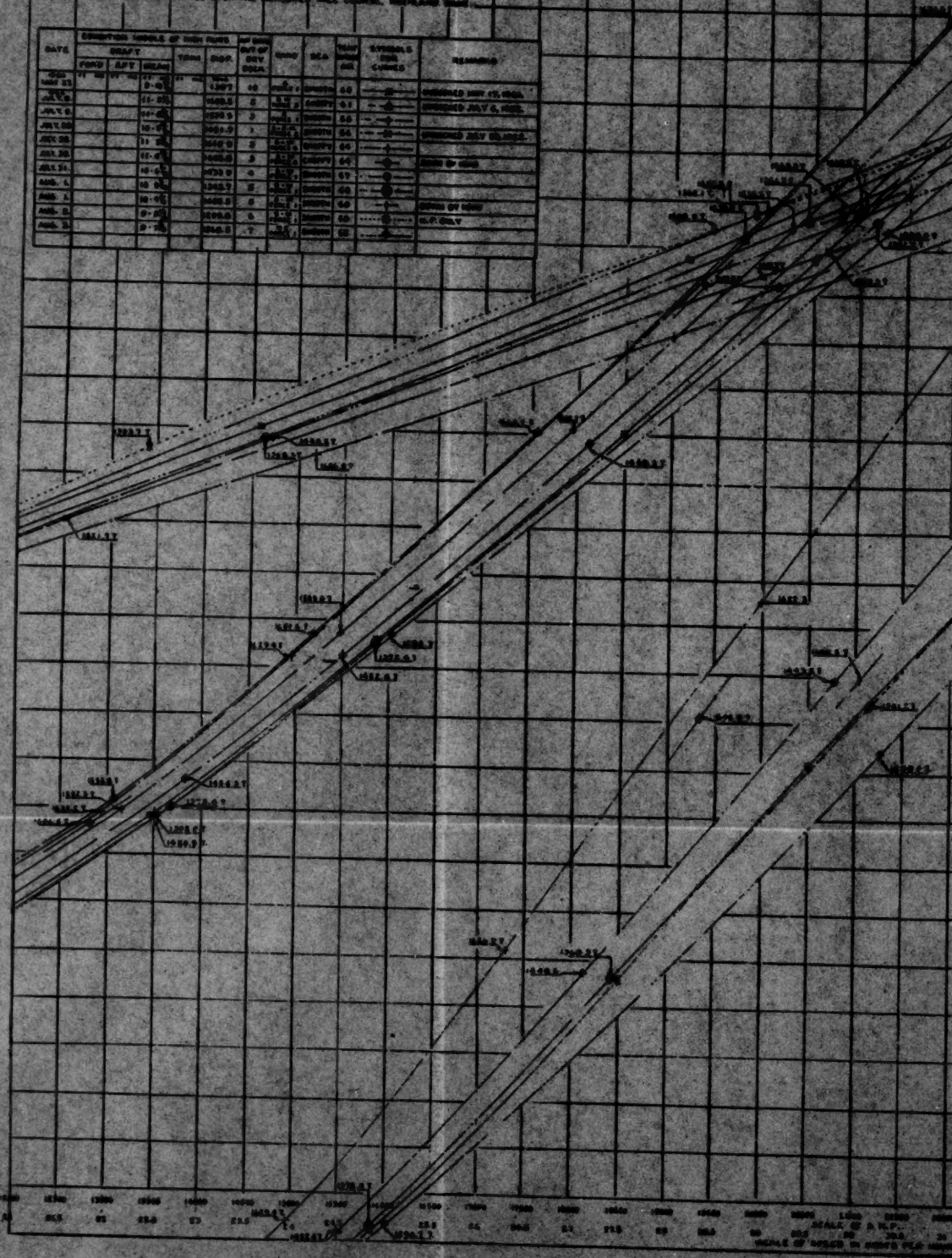
STANDARDIZATION TRIALS AT VARIOUS DISPLACEMENTS TO DETERMINE INCREASED RESISTANCE TO PROPULSION DUE TO INCREASE IN DISPLACEMENT

TRIALS CONDUCTED OVER THE MEASURED MILE COURSE, [illegible]

DATE	CONDITION WHEELS OF HIGH POINT			TRIM	DISP.	DEP. OUT OF DRY DOCK	SPEED	SEA	WIND VELOCITY AND DIR.	SYMBOLS USED IN CURVES	REMARKS
	DRAFT										
	FORE	AFT	MEAN								
1942	ft in	ft in	ft in	ft in	tons						
MAY 23			11-0		4387	10	[illegible]	SMOOTH	6.0		
JULY 6			11-0		4400.0	0	[illegible]	CHOPPY	0.1		DEPARTED MAY 23, 1942.
JULY 6			11-0		4400.0	0	[illegible]	CHOPPY	0.1		RETURNED JULY 6, 1942.
JULY 6			10-6		4298.9	3	[illegible]	SMOOTH	0.5		
JULY 28			10-6		4303.9	3	[illegible]	SMOOTH	0.4		
JULY 28			11-0		4348.0	2	[illegible]	CHOPPY	0.0		DEPARTED JULY 28, 1942.
JULY 30			11-0		4400.0	0	[illegible]	CHOPPY	0.0		
JULY 31			10-6		4330.0	4	[illegible]	SMOOTH	0.1		RETURNED JULY 31, 1942.
AUG. 1			10-0		4260.9	5	[illegible]	SMOOTH	0.0		
AUG. 1			10-0		4260.0	5	[illegible]	SMOOTH	0.0		
AUG. 21			9-2		4098.0	6	[illegible]	SMOOTH	0.0		DEPARTED AUG. 21, 1942.
AUG. 21			9-2		4060.0	-7	[illegible]	SMOOTH	0.0		RETURNED AUG. 21, 1942.
											H.P. ONLY



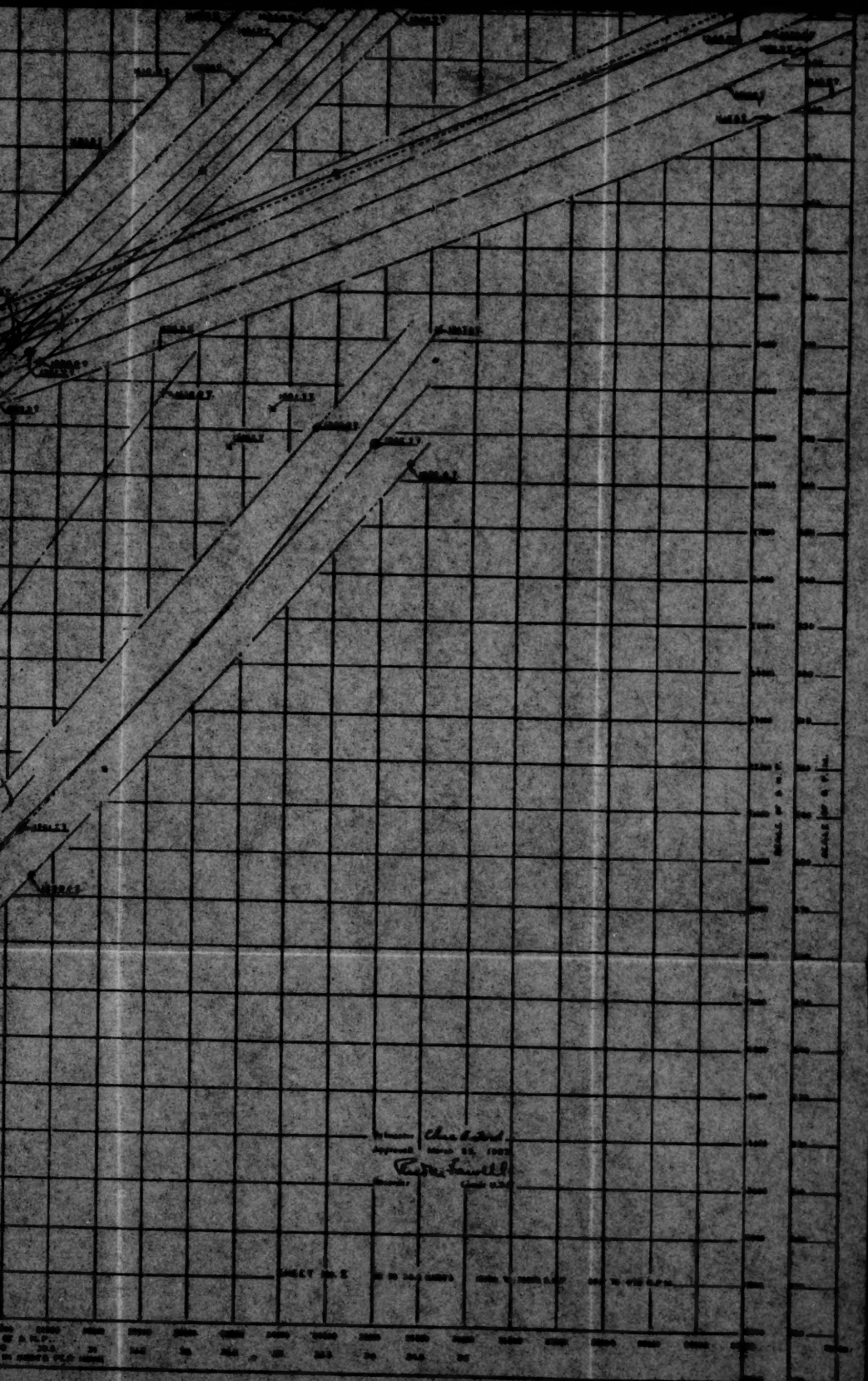
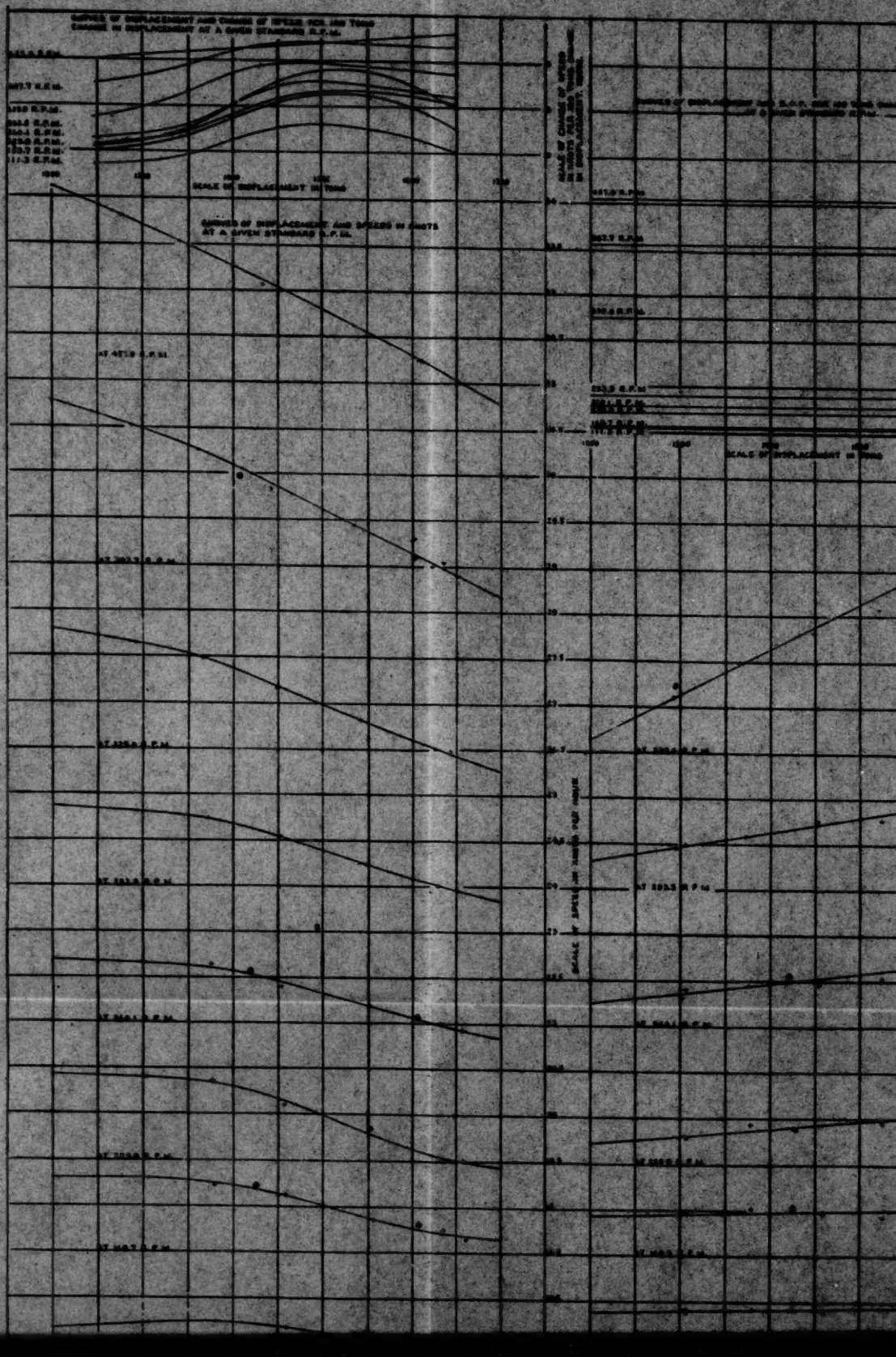
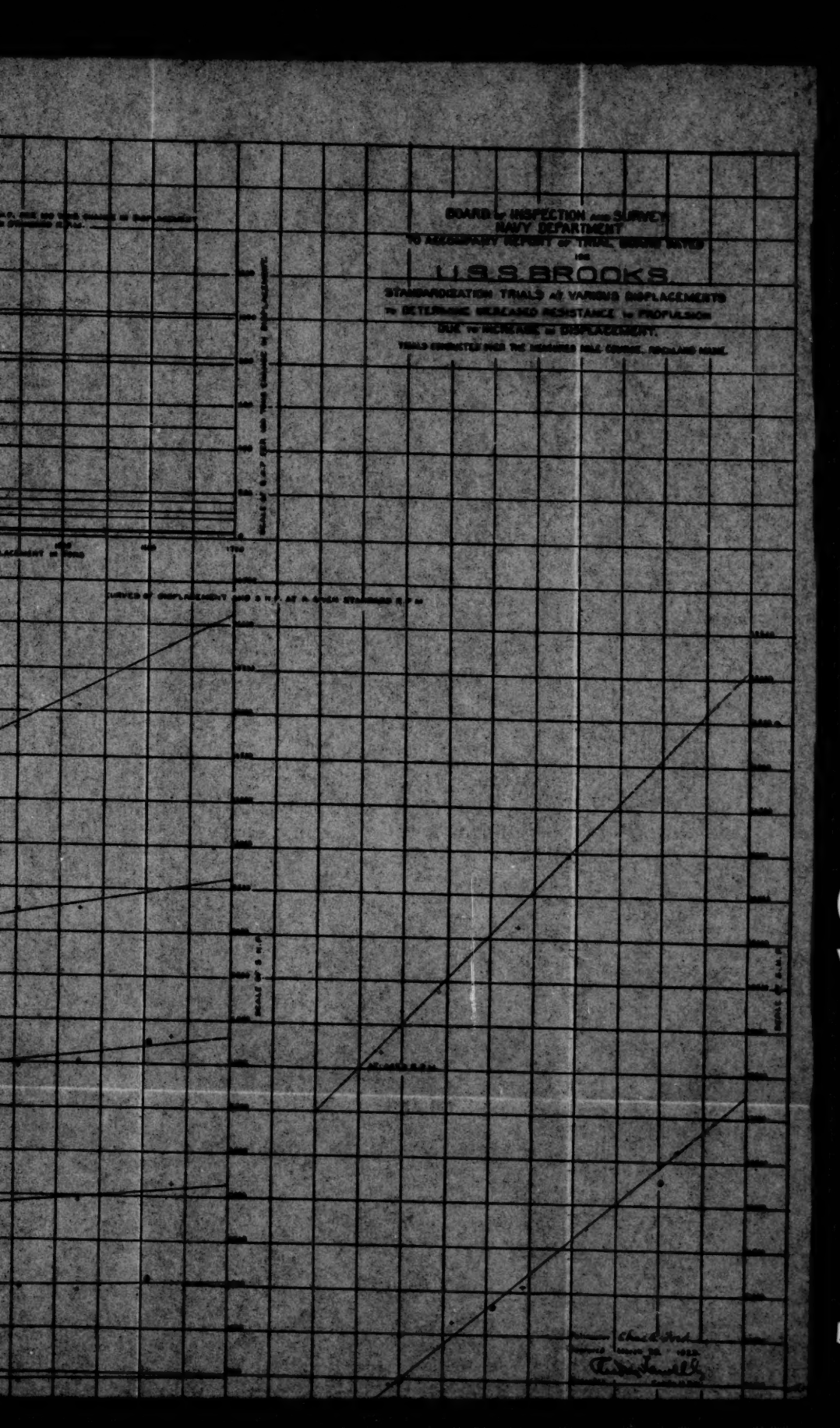


PLATE No 11A



BOARD for INSPECTION and SURVEY
NAVY DEPARTMENT
TO ACCOMPANY REPORT OF TRIAL, DATED [illegible]
ON
U.S.S. BROOKS.
STANDARDIZATION TRIALS at VARIOUS DISPLACEMENTS
to DETERMINE INCREASED RESISTANCE to PROPULSION
DUE TO INCREASE in DISPLACEMENT.
TRIALS CONDUCTED OVER THE MEASURED MILE COURSE, PENOBSCOT BAY, MAINE.



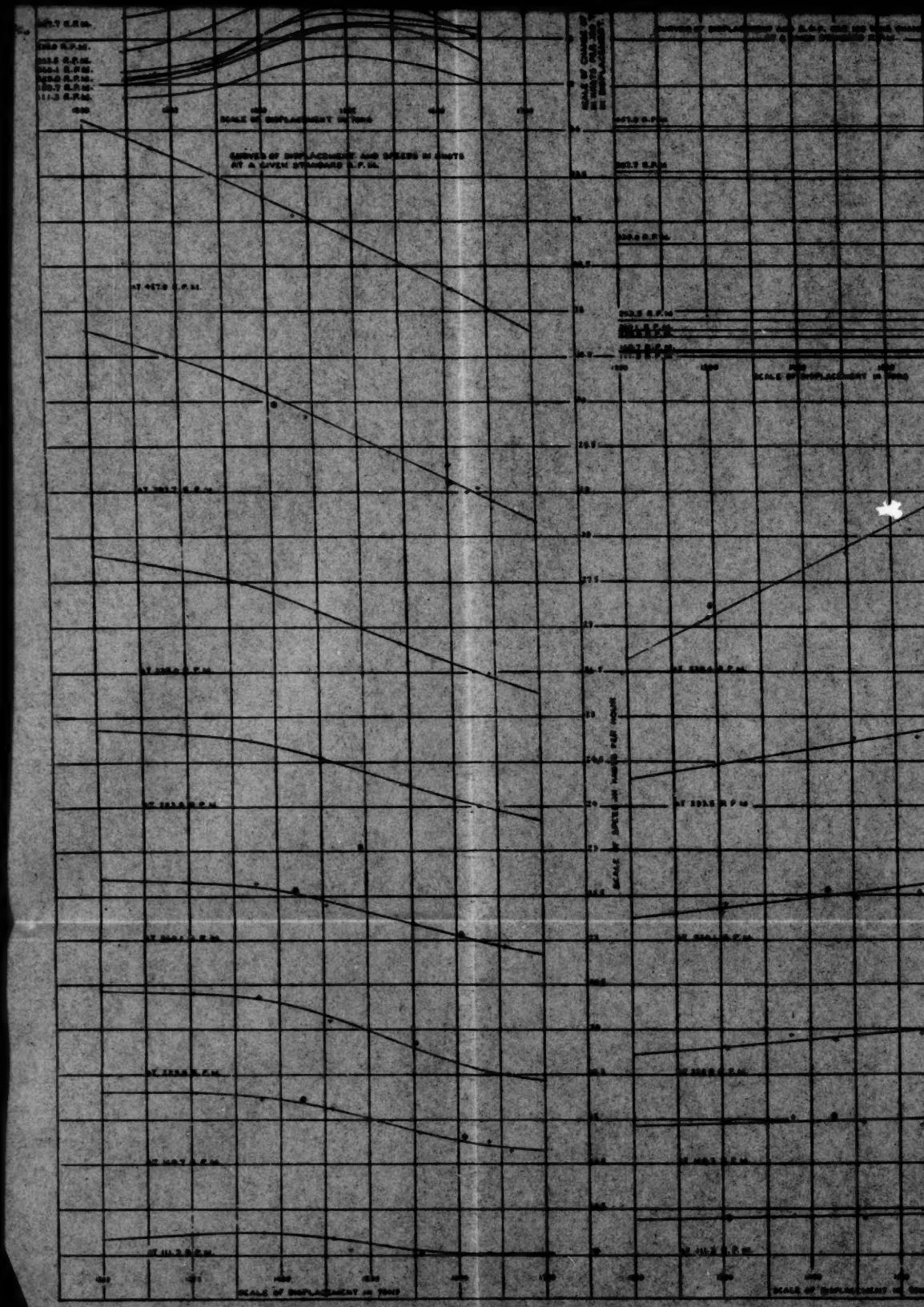

CURVES OF DISPLACEMENT AND SPEEDS IN KNOTS
AT A GIVEN STANDARD R.P.M.
SCALE OF DISPLACEMENT IN TONS
SCALE OF SPEED IN KNOTS PER HOUR

PLATE No 2.

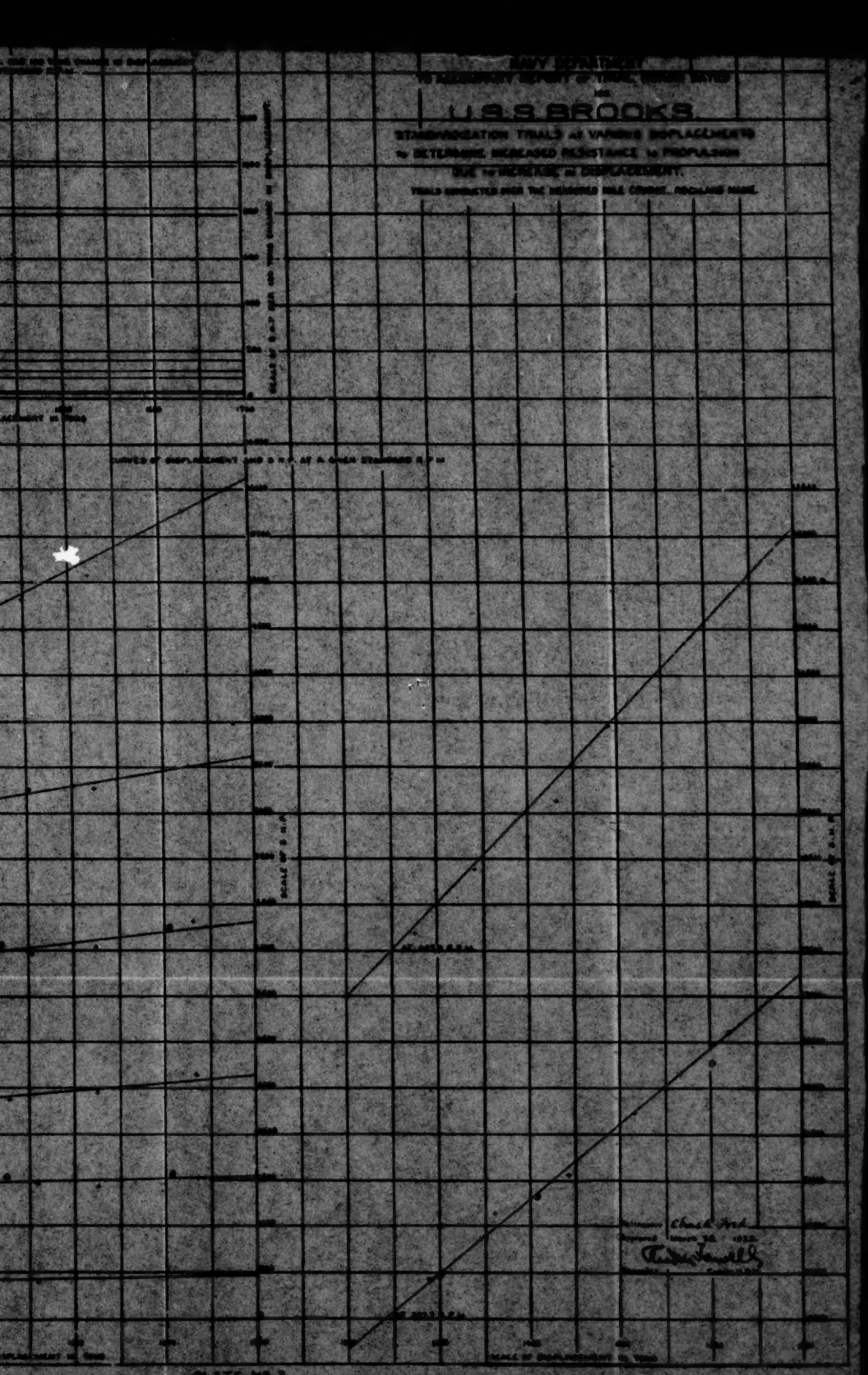

U.S.S. BROOKS
STANDARDIZATION TRIALS at VARIOUS DISPLACEMENTS
to DETERMINE INCREASED RESISTANCE to PROPULSION
DUE TO INCREASE in DISPLACEMENT.
TRIALS CONDUCTED OVER THE MEASURED MILE COURSE, PENOBSCOT BAY, MAINE.
CURVES OF DISPLACEMENT AND S.H.P. AT A GIVEN STANDARDIZED R.P.M.
SCALE OF S.H.P.
SCALE OF DISPLACEMENT IN TONS
Chief Naval Constructor
Approved January 30, 1922.

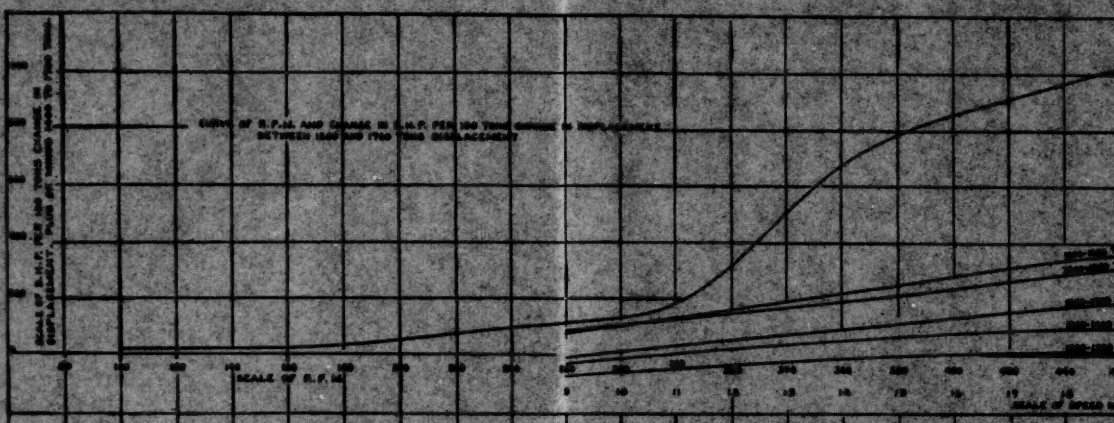
PLATE Nº 3.

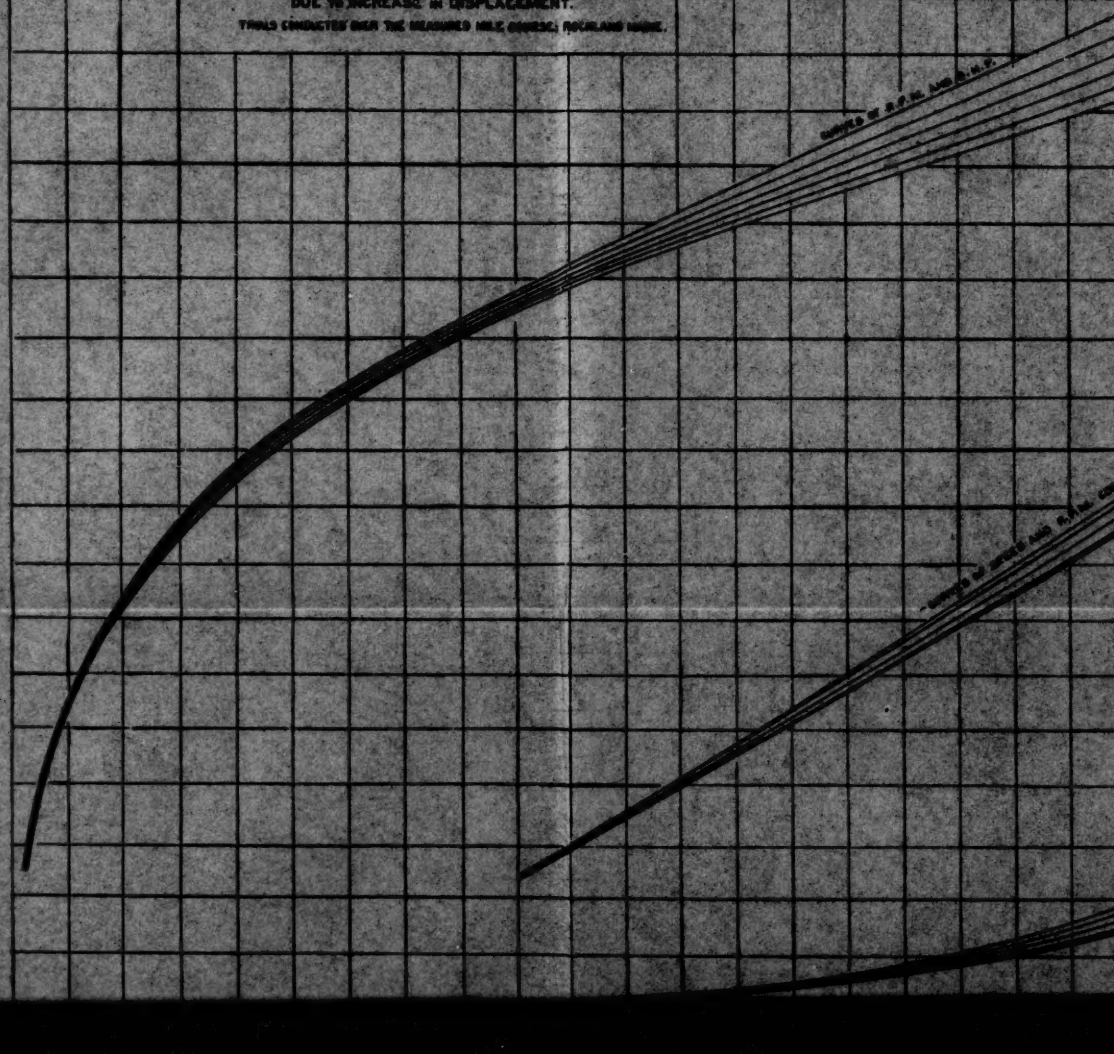
BOARD OF INSPECTION AND SURVEY
NAVY DEPARTMENT
TO ACCOMPANY REPORT OF TRIAL BOARD DATED
ON

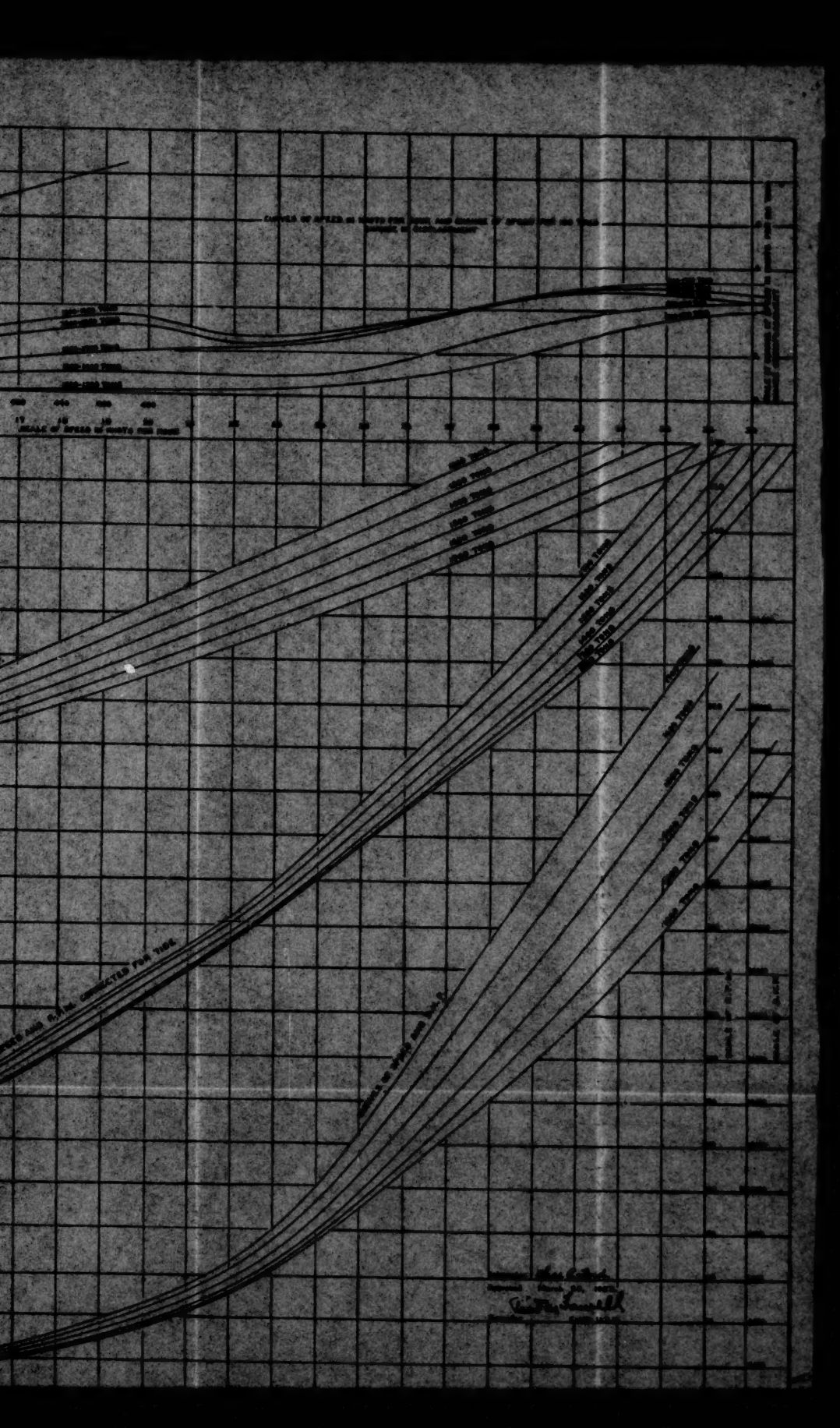
U.S.S. BROOKS

STANDARDIZATION TRIALS AT VARIOUS DISPLACEMENTS
TO DETERMINE INCREASED RESISTANCE TO PROPULSION
DUE TO INCREASE IN DISPLACEMENT.

TRIALS CONDUCTED OVER THE MEASURED MILE COURSE, ROCKLAND MAINE.



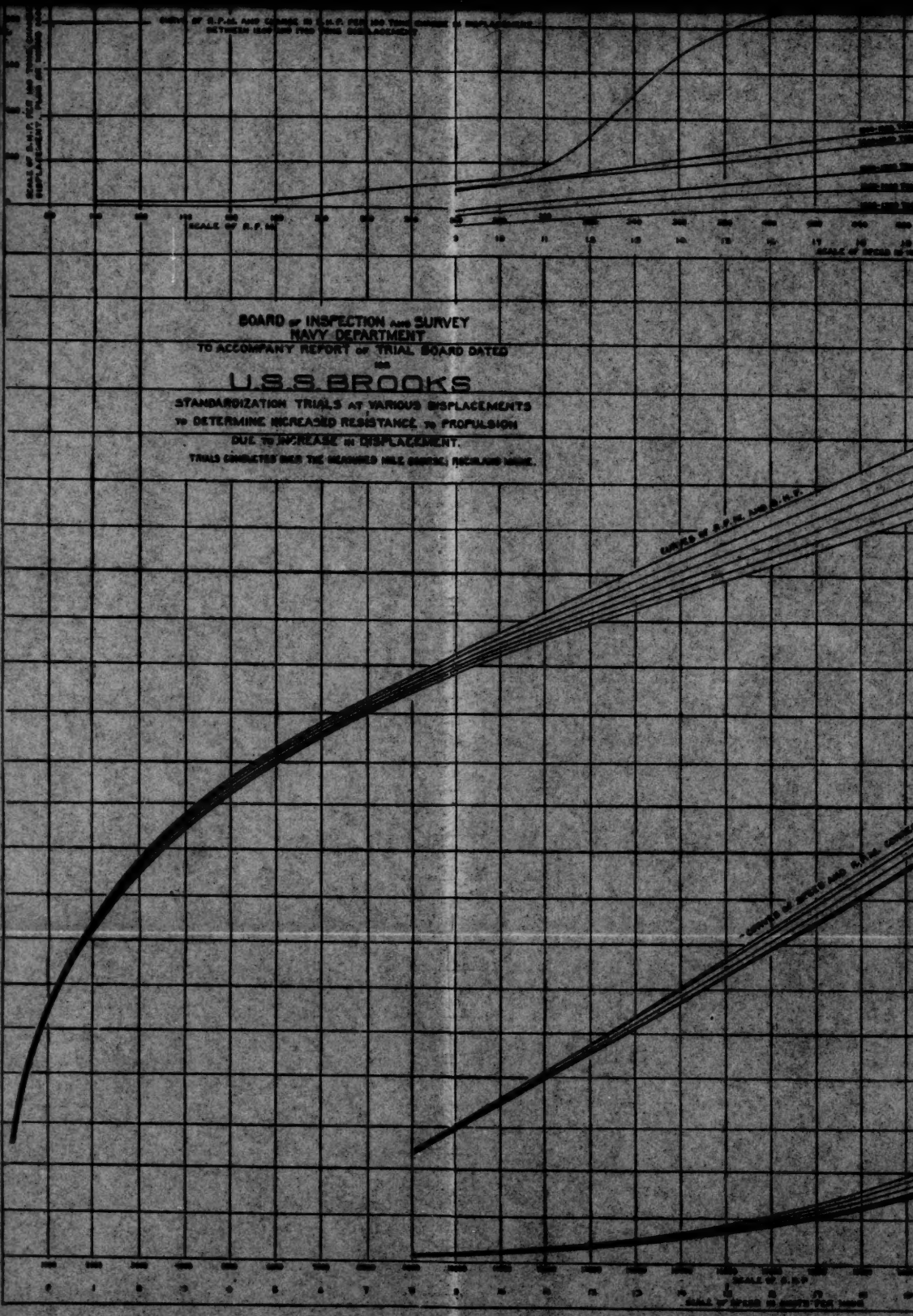




BOARD OF INSPECTION AND SURVEY
NAVY DEPARTMENT
TO ACCOMPANY REPORT OF TRIAL BOARD DATED
ON

U.S.S. BROOKS

STANDARDIZATION TRIALS AT VARIOUS DISPLACEMENTS
TO DETERMINE INCREASED RESISTANCE TO PROPULSION
DUE TO INCREASE IN DISPLACEMENT.
TRIALS CONDUCTED OVER THE MEASURED MILE COURSE, PENOBSCOT BAY, MAINE.



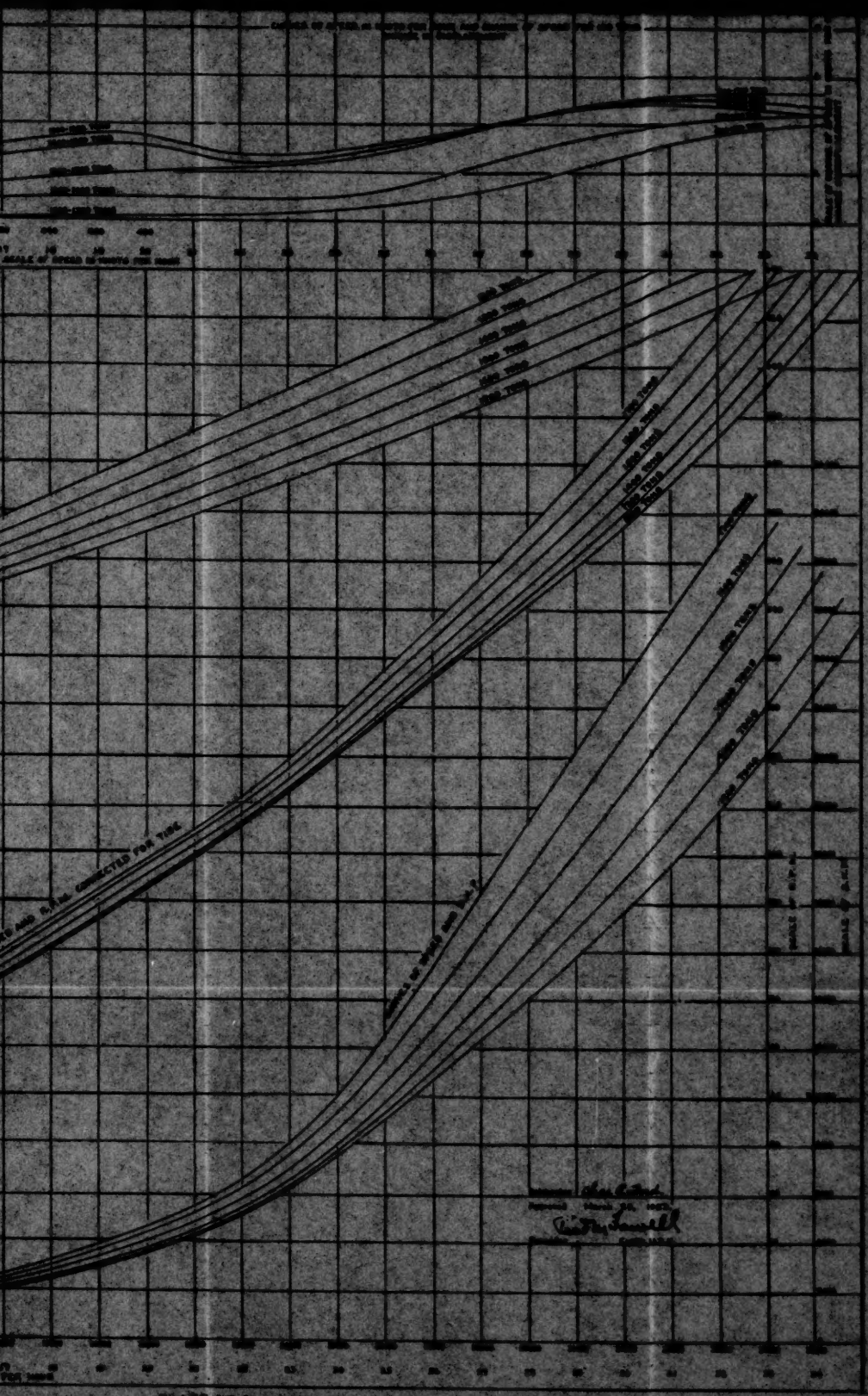


PLATE No 4.

extent if the displacement could be reduced a sufficient amount. This same shape of curve is shown on the curves of the *Putnam's* trials.

TABLE A.

Date of Trial	Displ. Middle of High Runs	Draft at the Beginning of Trial			Draft at the End of Trial			Remarks
		Forward ft. and in.	Aft ft. and in.	Mean ft. and in.	Forward ft. and in.	Aft ft. and in.	Mean ft. and in.	
1920								
May 27	1307.	9–9½	10–5	10–1¼	9–2⅝	10–4⅝	9–9⅝	
1922								
July 8	1608.2	11–9⅝	11–9¾	11–9¾	11–5 9/16	11–6 9/16	11–5 11/16	
July 9	1535.5	11–3¼	11–2⅞	11–3⅛	11–2 7/16	10–11 13/16	11–1 7/16	
July 28	1491.7	11–0¼	11–2⅛	11–1⅝	10–9 11/16	10–11 5/16	10–10¼	
July 30	1610.9	11–6⅛	11–9 7/16	11–8¼	...	...	...	
July 30	1602.8	12–5⅛	10–10⅜	11–7 11/16	12–2	10–10½	11–6¼	Down by the head
July 31	1433.9	10–9 1/16	10–9 7/16	10–9¾	10–7⅜	10–6 9/16	10–7¾	
August 1	1342.7	10–5⅜	10–2⅜	10–4¼	10–0 3/16	10–2 7/16	10–1 5/16	
August 1	1408.5	12–0 5/16	9–3⅜	10–7⅞	11–11	9–2	10–6½	Down by the head
August 2	1299.8	9–3 3/16	10–6 1/16	9–11	9–2¼	10–5½	9–9 1/16	
August 3	1268.5	9–3	10–6½	9–10½	8–10 7/16	10–6⅜	9–8⅜	

VACUUM.

By Commander O. L. Cox, U. S. N., Member.

For saturated water vapor or a mixture of vapor and liquid, the absolute pressure is a function of the temperature only. That is, for any given temperature the absolute pressure is a fixed definite amount, and any change in temperature causes a corresponding change in the absolute pressure.

If the vapor in a condenser were pure water vapor and did not contain any air or other non-condensable gases, the temperature of the vapor would give the absolute pressure and consequently the vacuum.

It is impossible to have air-free vapor in a condenser due to absorbed air in the feed water and air leaks throughout the system under vacuum, therefore, the temperature of the vapor in a condenser does not give the absolute pressure but only the partial pressure due to the water vapor present. The total pressure in the condenser is the sum of the partial pressures of water vapor, air and other gases present.

If the vacuum gauge fitted on the condenser is accurate, the difference between the gauge reading and the barometer reading is the absolute pressure in the condenser in inches of mercury, and the difference between such pressure and the pressure corresponding to the temperature is the partial pressure due to air and in a way is a measure of the air leakage. If there is any air present, the thermometer indicates a better vacuum than actually exists.

Ordinary water will absorb approximately 5 per cent of its volume of air and if this water is heated part of the contained air is driven off and when heated to about 200 degrees F. less than 1 per cent of the air is retained by the water. It, therefore, follows that the feed tank temperature should be carried as high as practicable for the feed pumps to handle the water. After the feed water passes through the feed heater, there is

usually fitted an air chamber to collect air and the escape valve on the air chamber should be kept partly open to allow the accumulated air to escape.

In the exhaust end of the turbine or in the exhaust trunk the ratio of air to steam is so small that the air has very little effect on the temperature so that a thermometer placed in either place will give practically the absolute pressure existing at this point.

It is important to keep all air possible out of condensers and to remove all air that unavoidably gets in, as the air surrounds the tubes and more or less insulates them and renders the condenser less efficient.

If the exhaust were entirely free from air, no air pump would be required and a small condensate pump would be all that was needed. Air pumps of sufficient size are fitted on all vessels to remove all normal air leakage and if with a given installation in good condition it is impossible to get proper vacuum with the air pump in proper condition, there is only one answer and that is *Air Leaks.*

As the vapor passes down through the condenser and is being condensed the proportion of air to vapor rapidly increases and at the point where there are equal weights of air and vapor a thermometer would show a temperature (corresponding to the partial pressure of vapor only) ten or twenty degrees lower than that corresponding to the total absolute pressure. The thermometer might show 79 degrees F. corresponding to a vacuum of 29 inches and the gauge show only 28 inches yet both are correct for what they measure, the gauge giving the actual vacuum in the condenser.

Under the usual conditions of condenser design the temperature of the circulating water rises from 15 to 20 degrees in producing the necessary cooling effect. To obtain economy in heating feed water the overboard discharge should leave the condenser at not more than 5 to 8 degrees below the temperature corresponding to the vacuum and the overboard discharge valve should be throttled down to give this condition. For

example, a vacuum of 28 inches corresponds to an absolute pressure temperature of about 101 degrees F. The maximum overboard discharge temperature would be about 96 degrees, so that with injection water of more than about 80 degrees it will be impossible to carry 28 inches of vacuum. Similarly, to obtain 29 inches of vacuum the injection temperature will have to be less than 60 degrees. These figures being based on the capacity of the circulating pumps or scoops usually fitted.

The following table shows the temperature corresponding to various absolute pressures in inches of mercury. The third column gives the corresponding vacuum for a 30 inch barometer. The sum of the absolute pressure and the vacuum equals the barometer reading. Conversely, the difference between the vacuum and the barometer reading gives the absolute pressure. For example, if the barometer reads 30 inches .75 and a correct vacuum gauge shows 28 inches .00 the absolute pressure is 2.75 inches of mercury, and if under similar operating conditions the barometer reads 29.75, the vacuum obtained would be 29.75—2.75=27 inches .00.

Absolute pressure inches of mercury	Temperature degrees F.	Corresponding Vacuum 30 inches Barometer
1.0	79.0	29.0
1.2	84.7	28.8
1.4	89.5	28.6
1.6	93.8	28.4
1.8	97.7	28.2
2.0	101.2	28.0
2.2	104.4	27.8
2.4	107.3	27.6
2.6	110.0	27.4
2.8	112.7	27.2
3.0	115.0	27.0
3.5	120.6	26.5
4.0	125.4	26.0
4.5	129.8	25.5
5.0	133.8	25.0

If the absolute pressure in pounds per square inch is desired, multiply the pressure in inches of mercury by .4912.

In reports of steaming trials of various vessels, failure to attain prescribed speed is frequently attributed to inadequate vacuum. This claim is generally correct but due consideration is not given to all factors entering into the problem of vacuum and the installation is criticised, whereas the trouble is due entirely to external causes.

The higher the vacuum for which a turbine is designed, the longer must be the low pressure rows of blades in order to accommodate the large volume of steam. It is general practice to design marine turbines for an absolute pressure of about 2.5 inches of mercury at the exhaust of the last row of blades when developing full power. Allowing a further drop in pressure to give velocity to the exhaust steam and overcome friction in the exhaust passages gives an absolute pressure at the top of the condenser of about 2.0 inches of mercury. The above conditions are based on the assumption that only the normal amount of air is present.

At full power, increasing the vacuum over the designed requirements will result in practically no increase in power, but will show that the condensing apparatus is in good condition and has practically no air leaks. This is due to the fact that blade and nozzle areas are so proportioned that the benefit would not extend back beyond the last two or three rows of blades. On the other hand, if the vacuum is less than required by the design, the effect extends back several stages and will result in more steam being required to develop the power.

When operating at less than full power, however, an increase in vacuum will have the effect of decreasing the steam consumption, as the increase in vacuum will extend back through several stages and more energy is available from the steam.

The external conditions which affect the vacuum are temperature of injection overboard discharge and indirectly the height of barometer.

If there were no air in the condenser, the air pump suction could be maintained at the temperature corresponding to the vacuum. The more air that is present, the lower the temperature of the air pump suction required in order to remove the air and the difference in these temperatures is a measure of air leakage.

Every means should be taken to see that the whole system under vacuum is absolutely tight. Air enters through labyrinth glands, flanges, valve bonnets and around valve stems. All joints should be gone over with asphaltum paint or shellac while the system is under vacuum and all glands on valve stems set up. Particular attention should be paid to pipe lines in the bilges and those subject to vibration. Every joint and valve in the system under vacuum should be thoroughly examined daily underway with a candle flame.

NOTES.

FUTURE OIL-ENGINE DEVELOPMENT.

A Summary of a Series of Important Papers on Modern Oil-Engine Progress Read Before the Verein Deutscher Ingenieure.

On June 29th a meeting, well attended by visitors from abroad, took place in Berlin, organized by the Verein Deutscher Ingenieure, under the chairmanship of Dr. Lippart Geheimer Baurat, of Nuremberg; the object of the meeting was the discussion of the present status of the Diesel engine and the problems connected with its development. The great importance of the Diesel engine as a prime mover was reflected at this meeting not only by the mere arrangement of it through the Verein Deutscher Ingenieure, but also by the contents of the chief lectures and the reports annexed to them. These reports, giving a true picture of the present status of our knowledge in this domain, will be of great interest to engineers in Great Britain.

THE DIESEL ENGINE OF THE PRESENT DAY.

Dr. Nägel, of the Dresden Technical High School, delivered the first lecture on the above subject. Starting from the paper which was delivered by the speaker in 1911 at a meeting of the Verein Deutscher Ingenieure in Breslau, on the status of the development of the Diesel engine at that time, he traced the further progress of this prime mover in four principal directions. These were discussed in detail by the aid of characteristic examples principally from the German industry. These four fundamental lines of development refer to:—

(1) The design of the engine and its adaptation to economy of material and high number of revolutions.

(2) The introduction of the two-cycle instead of the four-cycle principle.

(3) The change from compressed-air injection of fuel to the solid injection in the Diesel engine without a compressor.

(4) New measures for realizing an efficient combustion of heavy oils in the Diesel engine.

Referring to the first point, the paper discussed the new design of the engine frame; valuable experience has been gained with the high-speed submarine engine, which has been more or less utilized in the construction of all present-day Diesel engines. In the construction of the Diesel engine consideration must very often be given to cheap production in large numbers, this being for instance, of importance in the construction of the low-powered engine in series. So far as the large Diesel engine is concerned, economy of materials and space, whilst maintaining at the same time security in service and high thermal efficiency, is of paramount importance. By the aid of examples supplied by the firms of Maschinenfabrik Augsburg-Nürnberg, Gebr. Sulzer, of Winterthur and Ludwigshafen, Fried. Krupp A.-G., Germaniawerft, of Kiel, the Deutsche Werke A.-G., and Allgemeine Elektricitäts-Gesellschaft, of Berlin, among others, the features of construction of the engine in question were discussed, and finally the Michel motor was mentioned; this is built by the Michel-Motor-Gesellschaft m.b.H., of Kiel, and represents a new form of Diesel engine with a star-shaped arrangement of cylinders.

A 12,000 B.H.P. M.A.N. DOUBLE-ACTING ENGINE.

In adapting the two-cycle principle to large units, two-cycle valve-scavenging engines represented the starting point, followed by the port-scavenging two-cycle type, which, as marine engines, are expected to supply sufficiently large cylinder units. For the first time mention was made of the 12,000 H.P. two-cycle engine built at the Nuremberg works of the Maschinenfabrik Augsburg-Nürnberg for the German Admiralty in 1910 after much pioneer work; this engine was destroyed under the terms of the Peace Treaty after successful and highly promising trials. In the construction of valve-scavenging engines the Germaniawerft has made progress.

In changing over to the port-scavenging engine, Sulzers, of Winterthur and Ludwigshafen, have rendered pioneer service, and now build marine engines of this design up to 3,000 B.H.P. Excellent trial results with an entirely new engine built at the Ludwigshafen works of Messrs. Sulzer were detailed by the lecturer; by these results the two-cycle process has been placed, so far as the fuel consumption is concerned, on a level with the four-cycle engine for the first time. The Augsburg establishment of the Maschinenfabrik Augsburg-Nürnberg, after long trials, has recently developed a new two-cycle method, which is intended primarily for double-acting cylinders of the largest dimensions, and has given promising results on trial engines.

SOLID-INJECTION ENGINES.

The compressorless or solid-injection Diesel engine has become of great importance for the smaller powers, and has been developed along various lines. The omission of the compressor has become an economic necessity for the small Diesel engine, and has possibilities for large motors. The different designs and methods in question were systematically analysed by the author, and discussed by the aid of typical examples taken from the Motorenfabrik Deutz, the Germaniawerft, of Kiel, the Maschinenfabrik Augsburg-Nürnberg, and Sulzer Bros.

As to the utilization of heavy oils with high flash-points in the Diesel engine, inventive endeavors in Germany have received no economical stimulus during the past few years, because the price of the inland coal-tar oil has not been so much less than the price of the gas oil and lignite oil as to justify the use of coal-tar oil. Not until recently have heavy thick oils derived from petroleum been brought on the market from abroad which might resuscitate the work of inventors in this line. The only step forward besides the knowledge that two-cycle engines are especially well suited for the safe utilization of heavy oils without further measures is seen in the piston design developed by Krupp. During interesting trials made in the Essen works this has proved to be an excellent means for enabling coal-tar oils to be used exclusively, even under no-load conditions.

The lecturer finally mentioned the application of the Diesel engine to the drive of locomotives, and gave details of the manner in which this problem has been dealt with by the Linke-Hofmann-Lauchhamer-Werke, of Breslau, making use of the Lentz hydraulic gear.

Dr. Ing. W. Nusselt, of Karlsruhe, read a paper on heat transmission. The speaker mentioned tests which he had carried out with ball-shaped bombs for studying the cooling of hot gases. The heat passes into the cold wall of the bomb partly by heat conduction and partly by radiation of heat. In order to separate the whole transmission of heat into both elements, the lecturer has made use of bombs blackened on the inner surface on the one hand, and of those gilded inside on the other. The difference of delivery in both cases is proportional to the radiation of heat of the hot gases of combustion. It has been shown that the radiation of heat from hot gases follows the Stefan Boltzmann law of radiation; that is to say, it increases

with the fourth power of the absolute temperature, but it has been also shown that the amount of radiation by hot gases is only about 1-15th of the radiation by the absolute black body of the same temperature. The heat delivered by conduction depends on the temperature and on the pressure. By making use of the tests of Clerk with a gas-engine, the influence of the travel of the piston upon the heat given up was taken into consideration. Applying the new formulæ to the delivery of heat in the combustion engine, it will be seen that radiation does not play the part ascribed to it formerly, and that it varies considerably during the stroke. The assumed dependence of the co-efficient of heat transmission on the mean piston speed is certainly only approximate, and it is desirable that more exact tests should be made to give further data about the heat-transmission co-efficient in engines.

AIR-INJECTION AND COMPRESSORLESS ENGINES.

Dr. Ing. K. Neumann, of Hanover, reported on comparative tests of compressed-air atomization and compressorless service in internal-combustion engines. The speaker first remarked on the difficulties which are encountered by the change from air injection to mere pressure injection with heavy-oil engines, according to the Diesel method, and demonstrated by the aid of an example that airless injection under equal conditions requires extraordinarily high pump pressure and very small nozzle diameters. He further reported on trials with compressorless engines of different cylinder sizes at different numbers of revolutions. The combustion in these engines has been attained by the generation of eddies through the stroke of the working piston or with the aid of an ignition chamber, and the engines ran with petrol, with gas, or with coal-tar oil. All figures necessary for judging the working process were carefully measured on these tests.

In comparing the different methods of operation carried through, it is remarked that the oil engine of the new Institute of Internal Combustion Engines at the Technical High School of Hanover was used throughout the trials, and thereby any influence of the engine design on the results of the tests has been excluded. The results of the tests show the speed of combustion of the different oils within the engine, as well as all the phenomena influencing the diagrams and their reasons. It is shown that with high-pressure oil engines several ignitions will always occur, and, further, that it is of importance that the first partial ignition takes place in due time and at the desired place. By a calorimetrical investigation of the different engines the value of temperature is determined with air injection and compressorless operation during the periods of combustion and expansion, and in connection therewith the heat is measured which passes to the cooling water during those phases. It is found that the mean figure of transmission of heat will be greater with compressed air atomization than with compressorless injection, because in operating without a compressor at the time of ignition the eddy currents are not so strong.

GUIDING LINES FOR MASS PRODUCTION OF SMALL AND MEDIUM-POWERED OIL ENGINES.

A paper on this subject was read by Director F. Schulz, of Cologne-Deutz. Growing competition necessitates manufacturing engines in large quantities. In order to facilitate series production, it is necessary to maintain strict standardization, to use few sizes of cylinders, and to obtain intermediate output by changing the number of revolutions and the number of cylinders. All parts of different types must be standardized.

The series engine must be simple of construction in all its details. By low weight and short routine through the factory, production is increased. The centre of gravity of operations must be placed at the beginning of produc-

tion. Wrong measures in design and manufacturing methods which can be repaired in the construction of large engines will generally cause a catastrophe in mass manufacture.

FUELS AND THEIR COMBUSTION IN THE DIESEL ENGINE.

Chief Engineer Alt, of Kiel, spoke on this subject. He dealt with the most important chemical and physical features of the liquid combustibles as the basis for judging their suitability for the Diesel engine, accentuating the new efforts to adapt the engines especially for poor fuels. As the utilization of such oils is only possible in the Diesel engine, this has a special importance as compared with other machinery.

The speaker referred *in extenso* to the theories on the phenomenon of combustion within the Diesel engine, based on recent investigations. Our knowledge gained in this way has not a mere theoretical value, but is of great practical importance, enabling Diesel engines to be designed to burn any liquid fuel.

The former ideas about the phenomenon of combustion, (1) that the ignition of the combustible injected into the strongly heated air is preceded by vaporization to a certain degree, and (2) that the different behavior of combustibles in the engine must be ascribed to the differences of their behavior during the vaporization (opinions which we find generally adopted to-day), must, in the author's opinion, now be given up. On the contrary, it can be shown that the atomized combustible ignites directly; that is to say, ignition is not preceded by any vaporization worth mentioning. The behavior of combustibles during those phenomena is due mainly to their chemical (molecular) constitution, and is to be traced to the self-ignition temperature within the engine defined by that constitution. Combustion must further be carried through in such a way that no vaporization of combustible occurs, so far as possible, because the decompositions connected with it might cause the formation of carbon deposit and, therewith, poor combustion.

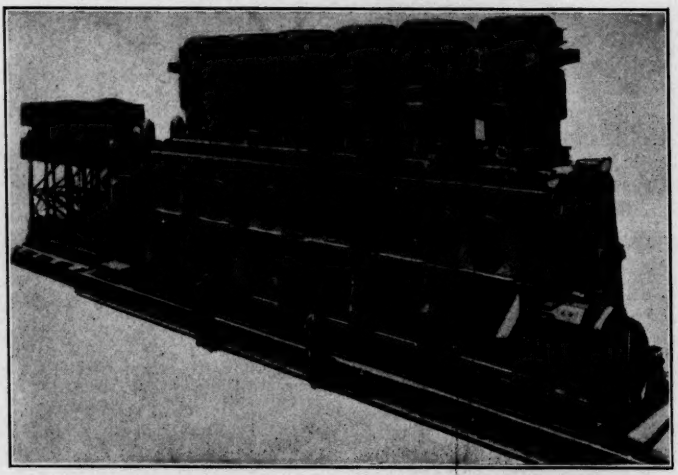
To prove the correctness of these new opinions, the speaker referred to the fact that the Diesel engine can be driven even with the most obstinate of all combustibles if conditions necessary for its due combustion be taken.

INCREASING THE SPECIFIC OUTPUT OF FOUR-CYCLE ENGINES.

In dealing with this subject, Dr. Riehm, of the M.A.N., Augsburg, said that development called for engines of greater and greater output. Overcoming the heat effects in the large cylinders gives rise, as is known, to the greatest difficulties. The heat stresses in the four-cycle engine are considerably lower than with the two-cycle type of equal dimensions, and investigation of the question, whether by increasing the effective pressure with the four-cycle engine we can attain greater output, is of special interest.

The mean effective pressure can be increased in a simple manner by introducing a greater quantity of combustible into the regular air charge of the working cylinder. The problem of burning this combustible in the given weight of air as completely as possible involves greater energy of injection, either by increasing the injection pressure or by feeding additional atomization air through a special valve. Tests of this method have shown, however, that, with increasing mean effective pressure, both the fuel consumption and the heat stresses increase rapidly. This method, therefore, can be taken into consideration only for temporary increase of output.

Better possibilities are opened by the method of precompressing the air drawn in, by means of which a greater air weight is admitted to the working cylinder. Combustion takes place approximately between the same limits of temperature and pressure as with the normal engine. The previously com-



A 12,000 B.H.P. Double-Acting Two-Cycle M.A.N. Engine in Course of Erection. It Has Since Been Destroyed.

pressed air is supplied by a blower. Tests have shown that an increase of output of 30 to 50 per cent can be obtained by this process without a considerable increase of the specific consumption of fuel. Heat stresses in the walls increase only to a small extent. Even more favorable conditions result if the energy of the exhaust gases is utilized for driving the precompressor plant; that is to say, by connecting the centrifugal compressor directly to the exhaust-gas turbine. This suggestion, which has been already tried with success with aeroplane engines for great heights, might offer possibilities with the Diesel engine and justify further tests.

VIBRATIONAL EFFECTS FROM OIL ENGINES.

Dr. Geiger, the inventor of the well-known torsiograph, read a paper on vibrational effects at a distance from internal-combustion engines. Among the long-distance effects from oil engines are vibration in dwelling-houses. These vibrations are caused by free forces, moments, etc. In order to ascertain the nature of these vibrations, in addition to the sensitive seismograph, the vibrograph is very suitable. Experience has shown that the sensibility of this measuring instrument, the writing pencil of which records vibrations on a continuous paper slip, goes as far as the sensibility of the human nerves; that is to say, it is adequate for the requirements of practice.

In order to prevent such long-distance actions, engines with excessive free mass forces must not be placed in the neighborhood of inhabited houses with several stories on a small area, because these fall very easily into resonance with the oscillation of the mass forces, and we know by experience that the resonance causes the greater part of disturbances. Attention must be paid to the building site as well, which must be reinforced by piles.

There are a number of expedients to remove vibrations: change of the crank angle, insulation, modification of number of revolutions, reinforcement of the shaken building, and compensating devices. These devices can be employed anywhere, and consist of creating mass forces by the aid of a device connected with the engine, which forces are of equal magnitude, but opposite to the mass forces arising from the engine.—"The Motor Ship," August, 1923.

A 12,000 B.H.P. DOUBLE-ACTING ENGINE

In the foregoing article reference is made to a 12,000 B.H.P. double-acting two-cycle Diesel engine, built at the Nürnberg works of the M.A.N., and which, it is stated, was destroyed under the terms of the Peace Treaty. An illustration of this motor in course of erection is given. We are not aware whether the engine was actually completed before its destruction was ordered, but it will be remembered that before the war a large double-acting plant was constructed at Nürnberg. During the course of the trials, an unfortunate disaster occurred, which wrecked the whole engine, and, presumably, the motor illustrated was constructed after that event. It has six cylinders, 850 millimeter bore with a stroke of 1,050 millimeters, which figures are interesting in comparison with the corresponding dimensions of the Burmeister and Wain four-cycle double-acting engine. This set has a diameter of 840 millimeters and a stroke of 1,400 millimeters, developing about 1,100 B.H.P. at 125 R.P.M., whereas the M.A.N. engine runs at 160 R.P.M.—a somewhat high speed for normal work. It will be noted that there are three separate scavenging pumps, driven off cranks on the extension of the crankshaft. Valve scavenging was employed, and the general design seems to be an adaptation of that of the double-acting engines of the motor ship *Fritz*, built by Blohm and Voss under license from the M.A.N.—"The Motor Ship," August, 1923.

COMPARATIVE DETAILS OF VARIOUS TYPES OF ENGINES.

In the accompanying table are given various features of two- and four-stroke engines built by Clyde manufacturers. This is extracted from the paper read by Dr. Mellanby at the summer meeting of the Institution of Mechanical Engineers, which was published in part in our last issue, when we were unable to include the table now printed. Various interesting differences between the engines are to be noted. For instance, it will be seen that Burmeister and Wain alone adopt spur gearing for the camshaft drive. There is an obvious difference of opinion between designers as to the suitability of taking the stresses through steel tie bolts or cast-iron columns,

Four-stroke Cycle.				Two-stroke Cycle.				
Design ...	Burmeister and Wain	Messrs. Harland and Wolff and Kincaid	Messrs. North British Diesel Engine Works, Ltd.	Beardmore-Toei	Sulzer	Messrs. North British Diesel Engine Works, Ltd.	Scott-Still	Camelhaird-Fullagar
Clyde Builders ...				Messrs. Beardmore	Messrs. Brown, Denny, Fairfield, Stephen, Weir		Messrs. Scott	Messrs. Brown, Rowan, Dunsmuir
Fuel Duration Control ...	When required	Various	No	Yes	Yes	Yes	Yes—Automatic	Yes
Crankshaft ...			Built	Solid-forged	Solid Forged	Semi-built	Semi-built	Solid Forged
Explosion Loads Thro' ...	Steel Tie-bolts		Steel Tie-bolts	Cast-iron Columns	Cast-iron Columns	Cast-iron Columns	Cast-iron Columns	Working Parts only (Steel)
Piston Cooling Medium ...	Fresh Water or Oil		Salt Water	Fresh Water	Salt Water	Fresh Water	Steam	Fresh Water
Piston Cooling Pipes ...	Telescopic		Jet System	Jet System	Jet System	Telescopic	None	Telescopic
Piston Cooling Pressure lb. per sq. in. ...	10-15		36	18	60	30	—	30
Starting Air Pressure lb. per sq. in. ...	350		350	500	1,600 at Bottles	350	Steam—140	400
Crosshead Slipper ...	Single or Double		Single	Single	Four Faces	Single	Four Faces	Double and with Side Faces
Camshaft Drive...	Spur		Skew	Skew	Skew	None	None	Skew and Bevel

Burmeister and Wain and the North British Co. preferring the former arrangement, and Beardmores, Sulzers, and Scotts the latter, which system is also employed in the North British double-acting engine. Moreover, there is no unanimity of view regarding the construction of the crank-shaft, and the various systems of cooling are a source of interest, particularly the employment of oil in certain Burmeister and Wain engines.

It should be added in this connection, however, that although in Harland and Wolff engines fresh water has usually been utilized for piston cooling, it has lately been common in the motors manufactured by Burmeister and Wain to adopt sea-water for this purpose and fresh-water for the cylinder jacket circulation. It is understood, however, that the employment of oil for piston cooling will be standardized in the future in Burmeister and Wain engines, even those of large size. In the Still engine, of course, as the underside of the piston acts as a steam engine, there is no need for any special cooling of the pistons, since the steam serves the necessary purpose. It will be seen from the table that the telescopic and jet systems for supplying the cooling medium to the pistons are equally favored. The starting air pressure of 1,000 pounds per square inch, even for the Sulzer engine, does not, of course, imply that the air is supplied to the cylinders at this pressure, but it is stored in the reservoirs at 1,000 pounds per square inch.—"Motor Ship," August, 1923.

HIGH-PRESSURE STEAM BOILERS.

Among the papers of particular interest presented at the seventeenth annual convention* of the Iron and Steel Electrical Engineers was that of Dr. D. S. Jacobus on high-pressure steam boilers. As an introduction Dr. Jacobus traced the history of high-pressure boilers from the time, over one hundred years ago, when Jacob Perkins built an apparatus to generate steam at several thousand pounds pressure.

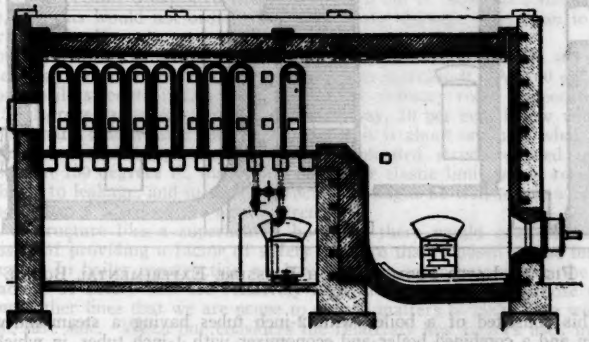


FIG. 1.—EXPERIMENTAL HIGH-PRESSURE BOILER DESIGNED FOR "SERIES" OPERATION.

He then described experiments made by the Babcock & Wilcox Co. about eight years ago with a boiler having a forced circulation and with no steam or water drum. This boiler (shown in Fig. 1) was purposely made of a form in which it was felt there might be certain difficulties, in order to

*Held Sept. 24 to 28, 1923 at Buffalo, N. Y.

search out the principles involved in such boilers. It was called a "series" boiler, because the water was forced continuously through a series of loops. Starting at the back, the first loops acted as an integral economizer, and the other loops, up to the separating tank (shown in dotted lines), as actual evaporating surfaces. The steam and water from the next to the last loop passed to the separating tank, from which the saturated steam rose to the final superheating loop next to the fire. The loops were made of 2-inch tubes with cores in the straight sections.

This boiler was operated at a maximum capacity of about 425 per cent of rating, based on the entire amount of heating surface of the economizer, boiler and superheater. The surface was not arranged in a way to give efficient results, the object of the test being to determine the features of circulation and the practicability of operating a boiler of this type.

Since this boiler had no water space and hence required 100 per cent reliability in the feed-pump system, it was decided to build a series boiler to be used in combination with a relatively small boiler having a steam and water drum. The installation is shown in Fig. 2.

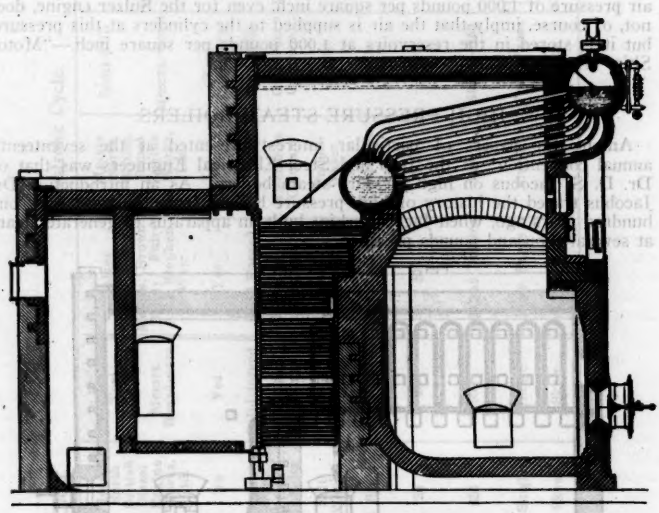


FIG. 2.—LATER FORM OF HIGH-PRESSURE EXPERIMENTAL BOILER.

This consisted of a boiler with 2-inch tubes having a steam-and-water drum and a combined boiler and economizer with 1-inch tubes, in which the heating surface was 3.7 times the heating surface of the boiler having a steam and water drum. For simplicity these will be called the upper and lower boilers respectively. There were 187 square feet of tube surface of the upper boiler, 51 square feet of superheater surface, and 685 square feet of boiler and economizer surface in the lower boiler, or 923 square feet in all. The boiler was oil-fired and operated at 600 pound gauge pressure. A connection fitted with a valve was run from the steam and water drum to the lowermost part of the lower boiler, and this connection was left open in

starting up so as to provide a local circulation. The unit was run, at times, at over 500 per cent of rating, based on the total surface, and all parts were found to operate in an entirely satisfactory way. The upper boiler operated at ratings as high as 1,500 per cent.

Experiments made with this boiler led to the final design of the 1,200 pound boilers now being built for the Weymouth and Calumet stations. Since this type of boiler has already been described in "Power,"* we may pass on to the following digest of Dr. Jacobus' comments on high-pressure boilers.

If it were possible to maintain absolutely tight surface condensers, it would be unnecessary to clean the interior of the boiler tubes and the problem of designing high-pressure boilers for power-plant service would be greatly simplified. Under present conditions there is always a chance of leakage at the condensers, so the problem both of cleaning the interior of the tubes and of removing any excess oxygen from the feed water must be faced in all cases.

The effect of temperature on the lowering of the elastic-limit of steel has been referred to by engineers as something that might give trouble in a high-pressure steam boiler. At a temperature of about 750 degrees F. the elastic limit may drop to half that obtained at ordinary room temperatures, and it is important that the effect of this falling off be understood. The temperature of the saturated steam corresponding to 1,200 pound pressure is somewhat less than 570 degrees F., and at this point the strength of the steel is higher than at ordinary room temperatures and the elastic limit is about the same or only slightly lower, so there need be no apprehension with respect to the boiler proper. The only part that is affected through being brought to a temperature that would cause any material difference in the strength or the elastic limit of the steel is the superheater. Of all parts of a boiler the superheater is the safest as far as its liability to injure anyone or to cause property damage is concerned, as the worst that might happen in case the superheater became overheated would be to burn out or blow out some of the tubes, and this would not ordinarily result in any damage other than to the superheater itself.

The effect of a rising temperature is first to increase the tensile strength of steel, say 10 to 20 per cent, and afterward to decrease it. At 700 degrees F. the tensile strength is about the same as at ordinary room temperatures. At 750 degrees the tensile strength falls off, say, 10 per cent below what it is at ordinary temperature, and the elastic limit is about one-half what it is at ordinary room temperatures. In a complicated structure used under pressure at 750 degrees F., the falling off in the elastic limit would result in a liability to leakage, and in such a structure it might be well to use a factor of more than 5 based on the tensile strength.

In a structure like a superheater, however, there would seem to be no necessity of providing a factor of safety of more than 5 based on the tensile strength. In boiler construction we have become so accustomed to the use of factors of safety that are relatively high in comparison with those used in some other lines that we are prone to overdo matters to make sure we are on the safe side. When one considers the relatively low factors used in the design of large guns and torpedo tubes carrying very high pressures, one feels a great deal safer when it comes to boiler design. Again, in the work we have done in designing and building oil stills and tanks for use at high pressures and a temperature of 825 degrees F., we made a careful study of the falling off of strength and elastic limit and of the practice in still design

*On May 29 "Power" described the 1,200-pound boiler at Calumet, while the July 10 issue described a similar boiler, at the Weymouth station.

that has been found to give safe results, and all this makes us more confident in feeling that the factors of safety we are using in boiler and superheater design are ample.

In the latest boilers for high-pressure and high-temperature superheated steam built by the Babcock & Wilcox Co., the superheater is placed above a series of tubes in a drop-leg and below the main body of tubes of the boiler. A superheater properly arranged in this position in a properly designed boiler gives a more constant degree of superheat at different ratings than if placed in the ordinary position above the boiler tubes, and permits a higher degree of superheat to be obtained with a given amount of superheating surface. Moreover, for the temperature of superheated steam now employed, the superheat obtained with a superheater of the sort is near enough constant for use without an attemperating device.

There is but little gain in economy in increasing the steam pressure to a higher point than 350 pounds per square inch without interstage heating of the steam; that is, withdrawing the steam from one stage, resuperheating it and returning it to the following stage. Interstage heating necessarily involves additional complication and added expense, and the possible difficulties in operation due to the additional complication have an important bearing on the use of the higher steam pressures.

Since the foregoing was written, a new development in the previously mentioned type of 1,200 pound boiler has been announced. This consists in moving the resuperheater from its original position enveloping the primary superheater in the space above the drop leg to a position above all the tubes. The purpose of this proposed change is to make possible the accurate regulation of reheat temperature by controlling the flow of gases past the reheater.—"Power," October 9, 1923.

GERMAN STEAM BOILER DESIGNED FOR 850 POUNDS PRESSURE.

By Bruno Schapira. *

In the course of the last twenty years, engineers in Germany, as well as in other countries, have come to realize the necessity of greatly increasing steam pressures if the steam-power station of small or moderate size is to compete successfully and permanently with gas and oil engines. Raising the steam pressure has particularly interesting possibilities in power plants where some of the steam is exhausted or bled for heating purposes. The higher the back pressure the greater is the relative advantage of high initial pressure.

The firm A. Borsig, in Berlin, is now constructing for its own use a steam-power station designed to carry a gauge pressure of about 850 pounds per square inch at the engine throttle. The 850-horsepower engine will operate with a back pressure of 140 pounds and drive an air compressor. The exhaust steam will flow into a heat accumulator, which in turn will supply the steam hammers of the shop. The boiler is designed to generate 15,500 pounds of steam an hour.

Fig. 1 shows the design of the boiler plant. The drums are forged from one piece of metal without welded or riveted joints. Fig. 2 is a shop view of one of these drums. To relieve any internal stresses, the drums are annealed after forging. Special plates, shown in Fig. 1, are provided to protect the upper drum from all contact with hot flue gases. There is sufficient space between these plates and the drum to permit inspection

*Consulting engineer, Vienna, Austria.

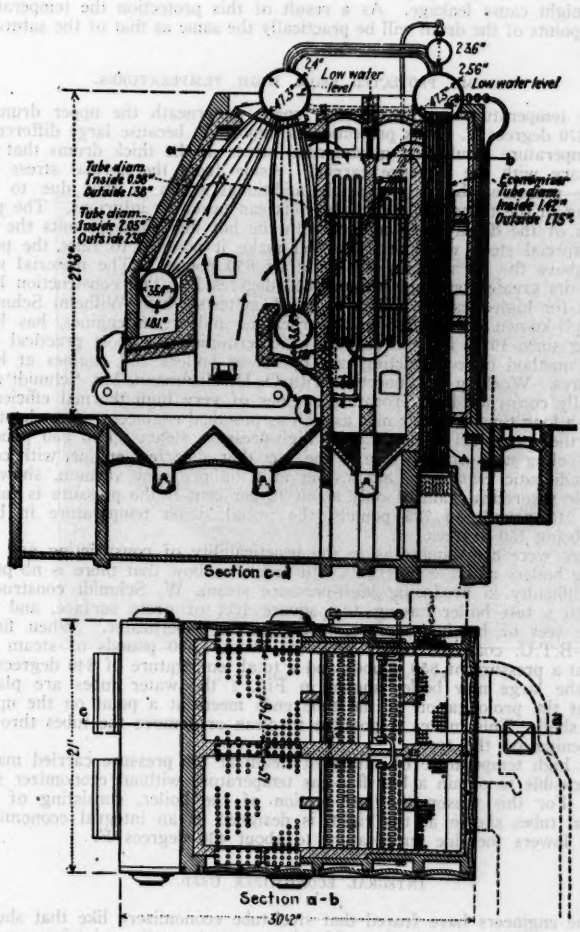


FIG. 1.—CROSS-SECTIONS OF BOILER AND ECONOMIZER.

of the tube connections at the drum while the boiler is in service. Sufficient clearance is allowed in the holes in the plate through which the tubes pass to allow normal heat expansion of the tubes but to prevent any vibration that might cause leakage. As a result of this protection the temperature at all points of the drum will be practically the same as that of the saturated steam.

DRUMS PROTECTED FROM HIGH TEMPERATURES.

The temperature in the isolating space underneath the upper drum is only 570 degrees F. This protection is important, because large differences of temperature would set up abnormal stress in the thick drums that are necessary with the pressure carried. Aside from the normal stress due to the pressure, the only possible additional stress is that due to expanding in the tubes while cold, but this can never be injurious. The protection of the drum from contact with the hot flue gases permits the use of a special steel, which in turn will make it possible to raise the pressure above the normal working point of 850 pounds. The material used shows its greatest strength around 540 degrees F. The construction here shown for high-pressure boilers is based on tests which Wilhelm Schmidt, the well-known constructor of superheaters and steam engines, has been making since 1910, for the purpose of determining the most practical and useful method of constructing and operating boilers and engines at high pressures. Working in connection with O. H. Hartmann, Mr. Schmidt successfully constructed reciprocating engines of very high thermal efficiency.

For a long time scientific men as well as practical engineers, were skeptical as to the successful application of high-pressure steam up to 850 pounds. This feeling was probably due to the fact that a perfect engine, with complete adiabatic expansion, exhausting at a 95 per cent vacuum, shows a possible theoretical gain of only about 12 per cent if the pressure is raised from 215 pounds to 725 pounds, the initial steam temperature in both cases being 750 degrees.

There were also doubts as to the practicability of constructing and operating boilers at these pressures. In order to show that there is no practical difficulty in producing high-pressure steam, W. Schmidt constructed in 1910 a test boiler having 10.8 square feet of grate surface, and 744 square feet of heating surface, including the superheater. When firing 13,500-B.T.U. coal, the boiler produced about 1,500 pounds of steam per hour at a pressure of 850 pounds and a total temperature of 840 degrees F.

In the large new boiler shown in Fig. 1 the water tubes are placed so that the prolongation of the tube ends meets at a point on the upper drum shell. This makes it possible to clean or remove the tubes through an opening on the top.

The high temperature of saturated steam at the pressure carried makes it impossible to obtain a low flue-gas temperature without economizer surface. For this reason the last section of the boiler, consisting of the vertical tubes shown at the right, is designed as an integral economizer, which lowers the flue temperature to about 430 degrees F.

INTEGRAL ECONOMIZER USED.

Some engineers have feared that steel-tube economizers like that shown would be subject to corrosion. Nothing prevents putting cast-iron economizers of the usual construction for low pressures behind the boiler. If the economizer is fed by a centrifugal pump there is only the feed-water pump to be regulated, which takes the heated feed water from the cast-iron economizer and delivers it to the high-pressure steel-tube economizer section. This arrangement makes it easy to degasify the feed water. As the

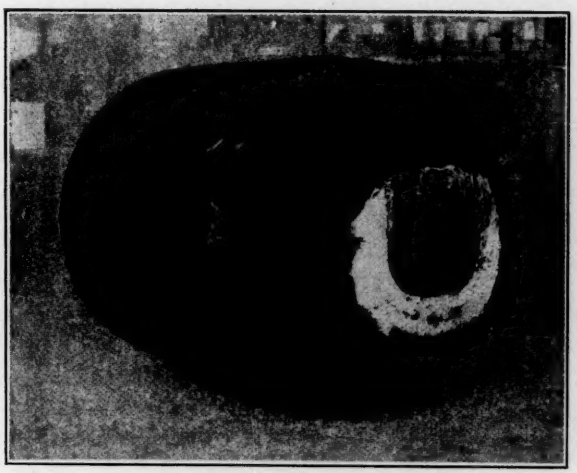


FIG. 2.—SHOP VIEW OF BOILER DRUM.

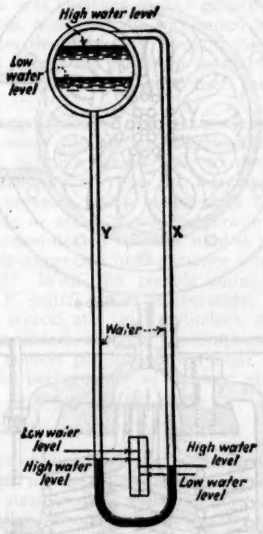


FIG. 3.—REMOTE-READING WATER-LEVEL GAUGE.

upper drums are protected from the flue gases, large variations in the water level are possible, and in this way sufficient space is obtained for large fluctuations in the feed.

The small test boiler was equipped with the usual fittings. The packing places of all fittings subjected to high pressure were made of nickel. Much difficulty was experienced with bursting gauge glasses. For this reason the remote-reading water-level gauge shown in Fig. 3 was constructed. It consists of a U-shaped tube, the lower portion of which is made of glass and contains mercury. The arm X connects with the steam space of the upper drum above the highest water level, while the arm Y is connected with the bottom of the drum. The condensation of steam in X keeps it full of water up to the point of connection to the drum, thus maintaining a constant hydrostatic head on this arm. The head on the arm Y is determined by the amount the water level lies below the drum connection of the arm X. The manometer scale at the bottom can therefore be graduated to read the difference in level of the two mercury columns in terms of water level in the upper drum. It is true that the movement of the mercury is small, necessitating fairly close reading of the manometer. This disadvantage, however, is more than compensated for by the fact that the gauge glass keeps cool and is not subjected to temperature strains.

LEAKAGE PROBLEM NOT SERIOUS.

No difficulties were experienced in making tight joints between the tubes of the test boiler and the drums. No leaks developed, nor were there other interruptions during operation. On several occasions slight leaks were observed in the economizer. These were easily remedied. The feed temperature ranged from 140 degrees F. to 175. After three years of service

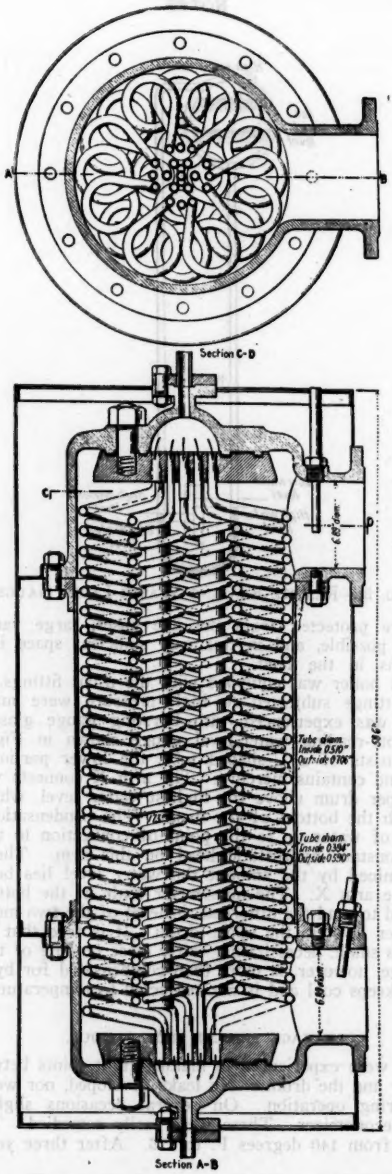


FIG. 4.—REHEATER OPERATED BY STEAM FROM BOILER.

the grate of the boiler was enlarged to 15.5 square feet, and by forced service and good draft the boiler output was raised to 2,960 pounds an hour. No harm resulted from this forcing. To date this boiler has operated for from 15,000 to 16,000 hours, and has been started at least 1,500 times. In this connection it is interesting to note that the feed water used was unusually dirty.

Equal success was experienced in the test made with high-pressure reciprocating engines. By using intermediate superheating and very great expansion in several steps, Schmidt found how to obtain practically complete expansion without condensation. For the intermediate superheating, saturated steam at boiler pressure and a temperature of 530 degrees was used. It was thereby possible to attain steam temperatures ranging from 430 to 480 degrees F. The steam-heated reheater used is shown in Fig. 4.

By using a quadruple-expansion high-pressure steam engine, the following results were obtained: With 780 pounds initial pressure, 95 per cent vacuum, 815 degrees F. initial steam temperature, 480 degrees steam temperature between the second and third cylinders, and 430 degrees between the third and fourth cylinders, a steam consumption (including intermediate superheating) of 5.8 pounds per horsepower-hour was obtained. This result, compared with the performance of the best piston steam engine today, corresponds to a heat saving of 22 per cent.

SUMMARY OF RESULTS.

The practical results obtained may be outlined as follows:

The production and use of high-pressure steam has been practically tested; high-pressure steam boilers and high-pressure steam engines can be constructed economically and operated with safety.

When operating condensing, the steam consumption per indicated horsepower-hour is about 5.8 pounds, as against 8.8 pounds for an operating pressure ranging from 215 to 285 pounds per square inch, the corresponding coal consumption per indicated horsepower being about 0.8 pound as against 1.1 to 1.2 pounds for the lower pressure.

The combination of power generation and heating, with the help of high-pressure steam engines, is in common use, as the back pressure, with such great initial pressure, is not a matter of so much importance as with steam engines operating at normal throttle pressures. The back pressure may, if necessary, be chosen as great as the initial pressure now commonly used. In this way exhaust steam may be used for processes for which it was formerly necessary to supply live steam direct from the boilers.

Existing power stations can also be changed over into high-pressure stations, either by altering the existing condensing steam engine for the use of high-pressure steam, or by putting a high-pressure plant ahead of the existing engines.—"Power," July 31, 1923.

HYDRAULICALLY PROPELLED MOTOR CRAFT.

THE APPLICATION OF THE HOTCHKISS SYSTEM TO TUGS.

It is an extremely interesting fact that, at a time when so much attention is being devoted to the question of improving the efficiency of ship's propelling machinery (mainly by the adoption of the internal-combustion motor), the problem of hydraulic propulsion should have been revived in a very practical form. It is very evident that, if results can be achieved with this method comparable with those obtained by the screw propeller in ordinary circumstances, the hydraulically propelled shallow-draught boat would show to considerable advantage over a similar craft in which stern wheels or tun-

nel sterns have to be employed to keep down the draught to the minimum limit required. If this be the case, a very wide field is opened out for the hydraulic system, and particularly in conjunction with oil engines in its application to motor tugs and commercial motor craft which have, perforce, to operate in shallow waters.

Very few accurate and certified figures are available regarding the exact performance of tunnel-stern and stern-wheel shallow-draught craft, but the details published below would seem to lead to the conclusion that the hydraulically propelled tug to which they refer is at least equal, and probably superior in efficiency and general suitability for its work, to a screw or stern-wheel-propelled craft built to meet the same conditions.

A DRAUGHT OF 11 INCHES.

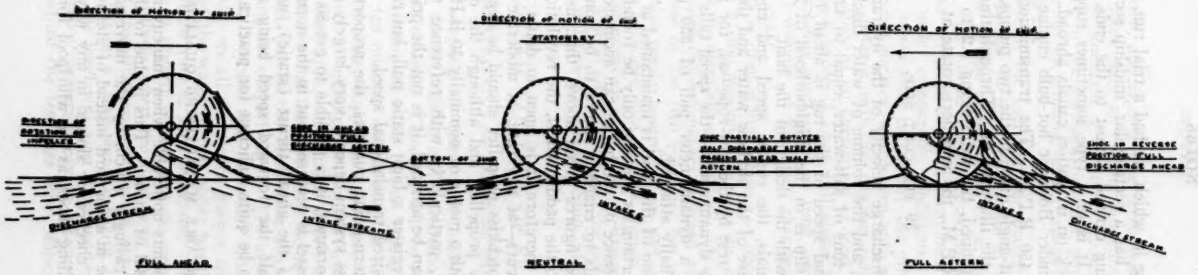
The vessel in question is a 40-foot tug designed by James Pollock, Sons, and Co., Ltd., for the Hotchkiss Propeller Co., Ltd., London, and from the results achieved with this demonstration boat the possibilities of the system used, if applied to larger vessels, may, perhaps, be gauged. The beam is 8 feet, the depth 2 feet 6 inches, and the draught light, with an unloaded displacement of 7.6 tons, is only 11 inches. When loaded with 4 tons of cargo the draught is 17 inches.

The tug is propelled on the Hotchkiss hydraulic principle, which has been successfully utilized on a number of different types of craft during the past two or three years. The Hotchkiss system is now becoming well known, but a few details of the principles employed may be briefly recapitulated. The engine drives an impeller which rotates within a drum casing. Between this casing and the impeller, and concentric with the drum, is a shoe which may be caused to rotate through an angle of about 70 degrees by means of the control lever. The purpose of this arrangement is to permit of speed control of the boat, and also to provide reversibility.

In normal operation, with the boat going ahead, water is drawn in through the inlet ports in the drum, and joins the vortex caused by the rotation of the impeller at some point between the center and the circumference. It is important to note in this connection that this point is chosen so that the rotational speed is about equal to the forward speed of the boat, or, in other words, to the speed of the water entering the impeller. On passing around the drum the water is accelerated, and the discharge takes place tangentially through the discharge ports in that drum, resulting in a side thrust on the drum, the thrust being a measure of the weight of water dealt with per second multiplied by the acceleration.

REVERSIBILITY.

With such an arrangement reversing is very simply accomplished. The shoe is rotated in the drum casing through the medium of the control lever. There is a reversing port in the drum, and when the boat is proceeding ahead this is covered by the shoe. The water is then necessarily discharged aft from the propellers, so that the boat moves forward. For reversal the main discharge port is closed by rotating the shoe, and the reversing port is uncovered. The water is then discharged forward and the boat goes astern. With the control lever in the central position, equal quantities of water are projected ahead and astern, and whilst the engine still runs on normal load the boat does not move in either direction. Between the limits of ahead and astern position of the control lever a very wide variation of speed can obviously be attained. Moreover—as in the standard arrangement—two pumps are provided, driven by the same engine—very fine maneuvering qualities are afforded, since, by independent control, one pump may be set for ahead and the other for astern movement.



SKETCHES ILLUSTRATING THE OPERATION OF THE HYDRAULIC PROPELLERS.

The demonstration tug in which we had a trial run, and which is now to be seen on the Thames, has a rectangular 'midship section with a perfectly flat bottom, the flat being carried almost to the ends and a short "swim" arranged at each end. It is of a type sometimes supplied by the builders for shallow-draught work on irrigation canals abroad.

A 30 B.H.P. two-cylinder Bolinder hot-bulb engine is installed, the full revolution speed being 450 R.P.M. The transmission is through a plain bevel gearing and a right-angle drive to the two pumps, one to port and one to starboard, as seen in the illustrations. Through this gearing the desired reduction in speed is attained, the ratio being 2.2 to 1, so that, with the engine running at 435 R.P.M., the impellers rotate at 198 R.P.M.

39 PER CENT SLIP.

The calculated main discharge velocity of the hydraulically propelled tug is 19.75 feet per second, and the volume of water discharged per second is 24.65 cubic feet or 7.04 tons of salt-water, equal to 42.2 tons of water per minute. The translational speed of the tug is about 7 knots, or 11.82 feet per second, so that the slip is in the neighborhood of 39 per cent, a figure which obviously varies with the design of the boat.

When running in canals, the engine speed and engine revolutions are reduced somewhat because of the shallow water and the consequently lower speed, and when held at rest by the tow-rope—as, for instance, when registering a static pull on the dynamometer—the speed falls to about 390 R.P.M. In these circumstances, a dynamometer pull of 550 pounds is registered, this being the figure actually attained.

Exactly what proportion of this pull is maintained at various translational speeds is at present uncertain, as this can only be found by extensive towing trials, whereby the resistance of the barge can accurately be gauged over a sufficient range of speeds to enable a curve to be constructed. It will be seen, however, from the figures given above, that, as in the case of the screw propeller, and also the paddle-wheel, the revolutions increase with the speed of the boat, and therefore it is open to doubt whether any serious falling off of thrust occurs—at any rate, at moderate translational speeds.

In this respect the Hotchkiss propeller should be regarded in exactly the same light as the screw propeller, and although the dynamometer pull of 550 pounds registered with a motor of nominally 30 B.H.P. is a performance which needs no excuse, particularly with reference to the very shallow draught with which it can be attained, it is not the prime object of this or any other propeller to register a high static pull, but rather to maintain its tractive effect at the desired translational speed.

With reference to astern-going qualities, the proportion of astern thrust obtained in the Hotchkiss system depends very largely on design. If astern speed is of primary importance, it is possible to get 80 per cent of the forward speed when the vessel is reversed, but in the normal design the thrust astern is roughly .56 of the ahead thrust (static), which results in considerably more than half the forward speed being available for astern going; this is thought to be quite sufficient for practical purposes, except in special cases.

REMARKABLE MANEUVERING QUALITIES.

One of the principal points noticeable when maneuvering the vessel is the suddenness of the reversal of thrust. This is due to the momentum of the water contained in the casings, which, by the movement of the reversing shoe, is allowed to escape in a forward instead of sternward direction. The engine and gear are not checked or slowed in any way by reversal or any movement of the controlling shoes, as the whirlpool formed in the drums is

not changed in direction, and is only interfered with to the extent that its peripheral discharge takes place at varying tangents to the circle, and since the torque required to maintain a constant revolution speed is the same whether the shoes are in the forward or after position, the engine is unaffected by maneuvering movements, and remains under constant load at constant revolution speed.

Note should be made of the fact that the maneuvering possibilities of a tug of this sort are similar to those obtainable in a twin-screw vessel, but the result is achieved by the employment of a single engine. During a somewhat prolonged trial of the tug, it certainly appeared to us that the "maneuverability" was better than could have been achieved by any other system of propulsion under comparable conditions. It is obviously quicker to move two control levers than to reverse the direction of rotation of a pair of directly reversible engines, whilst even with mechanical reverse gears it is scarcely wise to reverse with the engines running at full speed. In this hydraulic system there is no fluctuation of load, so that no compunction need be felt in changing suddenly from full ahead to full astern. The tug brought up in a remarkably short space of time, and on the Regent's Canal it was capable of all manner of evolution that would scarcely have been feasible with normal craft.

When running light in deep water a speed of about seven knots is maintained. Towing four 25-ton monkey-barges in the canal, the available tow-rope pull is sufficient to maintain a speed of 3½ miles per hour. When towing two partly loaded monkey-barges, a speed of 3½ miles an hour can be maintained with one of the two cylinders cut out, and a speed of five miles per hour with two partly loaded barges and a load of one ton in the tug. Finally, it may be added that Messrs. James Pollock, Sons and Co., Ltd., with considerable experience of shallow-draught tugs, are of opinion that there is no other system which can equal the performance of this system with similar draught and power.—"Motor Ship," July, 1923.

THE AUTOMATIC CONTROL OF FEED WATER.

A Description of the Apparatus as Fitted on the U. S. S. "Leviathan."

With the general adoption of water-tube boilers for vessels of large powers the necessity for a reliable and automatic system of feed water control at once became evident. This fact will be readily appreciated when it is stated that the amount of water in this type of boiler is seldom equal to more than five to ten minutes' evaporation at full output, and therefore the failure of the feed water supply to keep pace with the boiler evaporation even for a matter of minutes is fraught with serious consequences. There are, however, many other advantages which accrue from the practice of boiler feed control, not the least of these being the improved economy which may be obtained, due to the feed being always in proportion to the evaporation and the amount of fuel burnt. For instance, when a battery of boilers is fed by hand, it means that the water tender or the stoker has to keep an eye continually on all the water gauges of the series, and if one shows that the level of the water in that boiler has dropped, the check valve is slightly opened, and the rate of feed increased. Similarly, if the level has risen, the check valve is slightly closed, and the rate of feed reduced. This needs constant watchfulness when the load is in the least fluctuating. An alteration of level of less than an inch would not be noticed, though this, especially in a boiler having a large water capacity, is very considerable, and it is more often that the level will alter 2 inches before the rate of feed is altered and the necessary correction can be effected, especially if the man whose duty it is to adjust the feed has other duties to perform. As it is

far more dangerous to the boilers to let the water level get very much too low than too high, it is inevitable that the tendency should be to keep the water level well up in the boiler, as giving more margin of time for correction.

Now when the rate of feed is increased through the water level dropping, more water is being fed into the boiler than is being evaporated. This means a drop in pressure, which must be corrected by increasing the rate of firing. This means that the boiler is being forced for the time being, the heat of the chimney gases is increased and fuel is being consumed wastefully; or otherwise, that a larger quantity is being burnt for the same output than would be necessary if the rate of feed were always in exact proportion to the rate of evaporation. Though the quantity of fuel wasted each time the water level is pushed up, so to speak, is small, yet this may happen many times during the day's work, and these small quantities mount up to a very appreciable total in time. The fact that this waste is going on is not realized, as it is difficult to detect. With what is considered reasonably careful attention to the feed, there is a certain fuel consumption for a certain power, output. This is considered the normal efficiency of the boilers, and as long as it is not exceeded it is considered that all is well.

It is not possible with hand feeding to follow the demands of a boiler with a fluctuating load, even though the man in charge gives his undivided attention to the feed. With an efficient feed water regulator, however, it is possible to follow any change in water level with a maximum error of not more than ¼ inch either side of the water line, and cases are on record where such regulators have controlled the water level with a maximum variation of 5/16 inch. A natural corollary of the rate of feed inlet always being exactly proportioned to the evaporation demand, is that the circulation is greatly improved. It has been conclusively demonstrated that when feed water, in a greater quantity than is required, is fed into a boiler the circulation is at once impaired, or rendered more sluggish. It is also well known that a good circulation is essential in a steam generator and that a bad circulation results in the deposit of scale on plates and tubes being considerably increased, resulting in heat transmission losses, and much more work at the periodical examinations, which also have to be more frequent. Further, another important factor resulting from this control of feed water in accordance with steam required is the diminution of stresses often set up, due to rapid and unequal expansion and contraction caused by erratic feeding.

Having briefly outlined the advantages of automatic feed control, a description of a typical feed water regulator will be of interest. Fig. 1 shows a sectional view through a regulator and check valve as manufactured by the Aster Engineering Co., of Wembley, whilst Fig. 2 shows a regulator fitted to the drum of a water-tube boiler. The action of the regulator is as follows:—The valve A is similar in all respects to a standard feed check with the exception that it carries in its lower end a balance piston D, which is an easy working fit in the base of the check valve. Feed water enters the valve between the balance piston and the check valve and a small passage is so arranged that a small quantity of this feed water can pass to the underside of the balance piston by way of the special plug C. It will therefore be clear that if the outlet of the operating water pipe G is closed the pressure will accumulate in the space E equal to the pressure of the feed line, the balance piston D will therefore have no unbalanced thrust in either direction, the check valve A will therefore open and allow the water to pass to the boiler. If, however, the water from the space E is allowed to leak away by way of operating water pipe G and control valve K at a greater rate than it is supplied through the special plug C, no pressure will accumulate in the space E, and there will therefore be an unbalanced downward

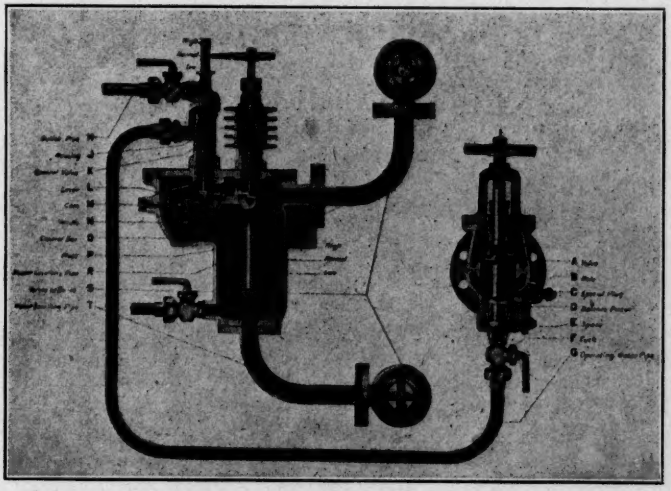


FIG. 1.—THE ASTER AUTOMATIC FEED REGULATOR AND CHECK VALVE.

thrust due to the feed line pressure acting on top of balance piston D, which in conjunction with the boiler pressure acting on top of the valve A will be sufficient to close the valve against the feed line pressure acting on the underside of the check valve and hold it closed till the operating water pipe G is again sealed and pressure is allowed to accumulate in the space E and so put the balance piston into equilibrium when the feed water will again pass to the boiler. From the foregoing it is obvious that a method of automatically sealing and unsealing the pipe G must be provided. This is done by means of the control valve K which is operated by the open float P contained in the control box O, which box is in communication with the boiler through the valves S. The level in the control box will therefore rise and

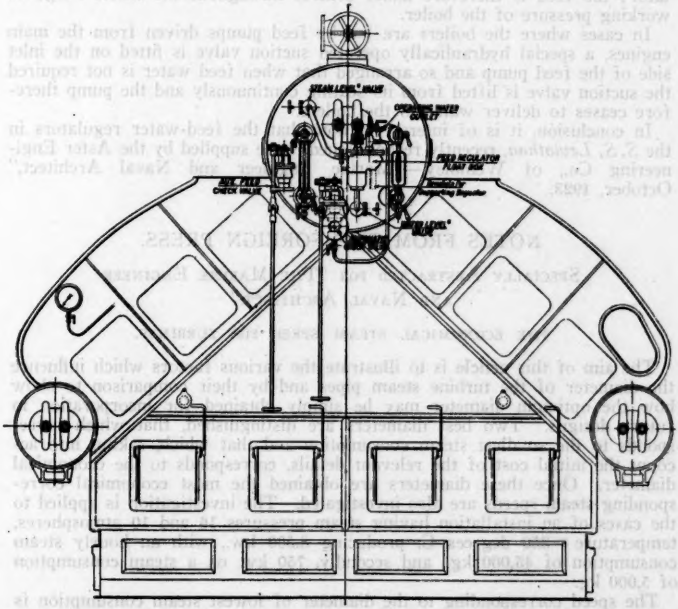


FIG. 2.—THE ASTER FEED REGULATOR AS FITTED TO A WATER-TUBE BOILER.

fall in sympathy with the water in the boiler, which movement is communicated to the float and thence by lever L to the control valve K. With the water at normal level the control valve will be so adjusted that it is just on the point of closing, therefore any fall in water level will have the effect of causing the control valve to close entirely the outlet orifice, thus sealing pipe G and allowing feed water to pass to the boiler as previously explained. The consequent rise in boiler level raises the float, causing the control valve to open, destroying the pressure under the balance piston of the check valve; the check valve is therefore returned to its seating and no more water can pass to the boiler. The operating water, after it has passed

the control valve, is led away by means of the outlet pipe H and returned to the hot well or feed pump suction without loss. Should it at any time be necessary to revert to hand feeding, it is only necessary to close the cock F, which instantly converts the automatic check valve into an ordinary hand-regulated check. It will be noticed the float is of the open type and is maintained full of water by the small air condenser fitted to the cover directly over the float. This air condenser is also used to house the adjusting spindle by means of which the position of the float relative to the control valve can be raised or lowered, thus providing a means of varying the water level if desired whilst the regulator is in action.

From the above description it will be observed that variations in temperature and pressure do not in any way affect the working of the regulator; the feed is therefore under control throughout the whole range of working pressure of the boiler.

In cases where the boilers are fed by feed pumps driven from the main engines, a special hydraulically operated suction valve is fitted on the inlet side of the feed pump and so arranged that when feed water is not required the suction valve is lifted from its seating continuously and the pump therefore ceases to deliver water to the boilers.

In conclusion, it is of interest to note that the feed-water regulators in the *S. S. Leviathan*, recently reconditioned, were supplied by the Aster Engineering Co., of Wembley.—"Marine Engineer and Naval Architect," October, 1923.

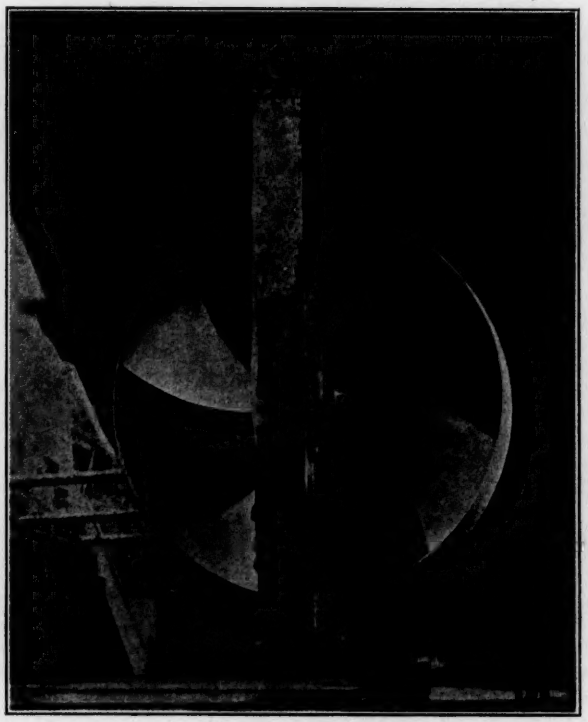
NOTES FROM THE FOREIGN PRESS.

Specially Abstracted for "The Marine Engineer and Naval Architect."

THE ECONOMICAL STEAM SPEED FOR TURBINES.

The aim of this article is to illustrate the various factors which influence the diameter of the turbine steam pipes and by their comparison to show how the optimum diameter may be simply obtained for incorporation in future designs. Two best diameters are distinguished, that which corresponds to the smallest steam consumption and that which, taking into account the initial cost of the relevant details, corresponds to the economical diameter. Once these diameters are obtained the most economical corresponding steam speeds are also investigated. The investigation is applied to the cases of an installation having steam pressures 16 and 10 atmospheres, temperature = 350 degrees C. producing 8,500 kw., with an hourly steam consumption of 45,000 kg., and secondly, 750 kw. on a steam consumption of 5,000 kg.

The speed corresponding to the diameter of lowest steam consumption is lower than that corresponding to the most economic diameter. These speeds are much less than those usually recommended, *i.e.*, 80 to 100 m./sec., and are less than 48 m./sec. The best steam speed is higher and more economical, the higher the steam temperature the lower the pressure, and the smaller the turbine. For a given turbine steam pressure and temperature the economical speed is higher the less the resistance caused by valves and water separators. This fact is clearly shown in a detailed example and brings out the superiority of a new type of valve, called the Kaswa valve, over the ordinary type for steam turbine work. With this valve for large turbines having separators the best speed varies from 33 to 35 m./sec., while with the ordinary valve the best speed is much lower and hence less efficient, and varies from 19 to 25 m./sec.—*K. Schmidt, Schiffbau*, May 30, 1923, p. 554; 3 pp.; 3 ff.—"Marine Engineer and Naval Architect," August, 1923.



VIEW OF THE PROPELLER AS FITTED TO THE SALVAGE VESSEL "RINGDOVE'S AID."

THE GILL SHROUDED PROPELLER.

Some Notes on Its Inception, Characteristics and Application.

The idea that a screw propeller fitted with a shroud should possess advantages over the open screw has occurred to many investigators during more than sixty years past. Attempts to realize these advantages did not, however, meet with conspicuous success, for a screw propeller fitted with a shroud was invariably found to be inefficient as compared with the open screw. Some sixteen years ago the designer of the Gill propeller was induced, by experience of certain propeller troubles, to consider the apparent possibilities of the shrouded type of screw and to attempt an analysis of its defects. The subject was worked at intermittently during a period of over ten years without any definite advance on the results obtained by former enquirers. When investigating during 1915 the then unconnected subject of flow in nozzles and bellmouths, however, it occurred to the designer that a definite connection exists between the design of particular forms of nozzles giving high coefficients of discharge and that of a shroud correctly applicable to the flow conditions through certain screw propellers. This connection was further supported by detailed consideration of Mr. R. E. Froude's theoretical conception of an imaginary "actuator" and tube.

It received additional justification during 1919 by deductions based on certain experimental results which were published by Sir Charles Parsons and Mr. S. S. Cook. A new series of experiments was therefore commenced with various types of nozzle-shaped shrouds and with model screw propellers, treated both independently and together, with the result that, in 1920, a screw propeller, having axially increasing pitch and a nozzle-shaped shroud of particular form and proportions, was first produced. This propeller gave practical results definitely better than those obtained with certain good typical screws of the open type. It was followed during 1921 by the production of a number of gradually improved screws of the same class and various proportions. During 1922 a consistent series of excellent results was obtained with such propellers, fitted to a great variety of craft and ranging from 1 foot to 6 feet diameter. By the end of the first quarter of 1923 a sufficient quantity of full-scale trial results had been collected to permit of an analysis of the leading proportions which resulted from the successful combination of theory and practice. This analysis has rendered possible the standardization of the features of design to the extent that all ordinary cases in practice can be met from a set of type drawings and patterns.

DESCRIPTION OF THE PROPELLER.

The Gill propeller consists of a screw, having two or more blades, which are mounted at their inner ends on a boss, or hub, and fixed at their outer ends to a shroud of special form, surrounding the screw. The shroud contracts in nozzle shape from the forward to the after end of the propeller, and the working pitch of the blades increases axially in the same direction. The blades and the shroud are designed to produce regular acceleration in the stream acted upon, according to the recognized laws of fluid flow. The boss, blades and shroud constitute a complete propeller, and rotate as a unit.

A typical open propeller with a shroud fitted to it would not constitute a satisfactory shrouded propeller, even if the shroud were of good form. In the first place, it would overload the engine to which it had hitherto been suited, and, in the second place, it would lose much of its efficiency. The theoretical considerations affecting the design of the Gill propeller are somewhat complicated, for they have to take into account a great variety of factors. In practice, however, this propeller is made from 10 to 15 per cent

smaller in its determining proportions than is an open screw for the same installation. This is chiefly because all the water which enters a Gill propeller is compelled to go through it in the direction required to produce reaction or thrust, and the propeller, therefore, takes on a more direct load for its size than does the normal type of screw. The same feature serves to furnish a reason for the fact that this propeller gives effective results, even under conditions of obstructed and oblique inflow, in addition to partly accounting for its freedom from vibration, and explaining the remarkable initial acceleration displayed by any vessel to which it is fitted.

The distinguishing feature of this screw, in causing all the water acted upon to be thrust directly astern, is one of great importance, for it is obviously impossible to develop a full forward reaction from water which is thrown out divergently.

SLIP.

This introduces the much misunderstood subject of "slip," and it is well to note that a propeller, whose advance through a solid "nut" would be from one and a quarter to one and one-third times that of the actual forward advance which it makes through the water, is, generally speaking, more efficient than propellers with less so-called "slip." A further notable feature of this propeller is that it produces better results when working under very shallow immersion than do open screws under like conditions, being distinctly less liable to "break surface" than are screws of the open type and being less affected by the air taken down when this does occur. Among objections which have been advanced, the first is usually the increased frictional surface due to the shroud. This would be valid except for the fact that considerably more useful effect is produced by the shroud than is lost by its surface friction. The next objection frequently raised is that the shroud should preferably be fixed, with the propeller rotating within it. Such an arrangement would be unsound hydraulically on account of excessive eddy losses, and unsound mechanically by reason of the difficulty of maintaining alignment in service.

Another objection put forward is, that solid objects may become wedged in the propeller, but it is found in practice that, if too large to go through, they are thrown off. Further and more far-fetched objections are the centrifugal tendency to bursting of the shroud and the possibility of its acting as a "drogue." These suppositions lose sight of the fact that, owing to the velocity of flow, the internal pressure is much lower than that outside the shroud, while in true nozzle flow the pressure normal to the internal surface of the nozzle has a zero value.

RECENT RESULTS.

A recent case in which one of these propellers, of medium size, viz., 10 feet 6 inches diameter, had been applied to a steam-driven vessel of 1,200 I.H.P. with very good results, is that of the salvage ship *Ringdove's Aid*, belonging to the Falmouth Docks & Engineering Company. This vessel is a composite-built ex-naval sloop, almost identical with H.M.S. *Greyhound*, made famous by Mr. W. Froude's classic towing experiments, and very suitable for salvage work. A new and specially built set of compound engines, by J. Samuel White & Co. Ltd., has recently been installed, and ample propulsive data being available for this vessel, it was calculated that a sea speed of 11 knots was to be expected with the 1,200 I.H.P. and a good four-bladed propeller running at 120 R.P.M. In view of the successful results obtained with Gill propellers as applied to smaller craft in the service of the same company, however, it was decided to fit one to this salvage ship, and the screw was cast in iron at the works of Cox & Co. (Engineers)

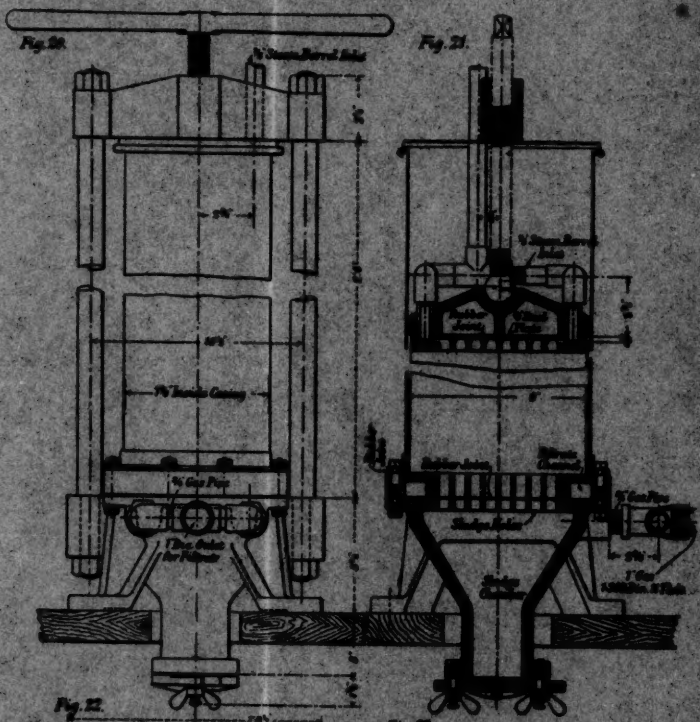


Fig. 20.

Fig. 21.

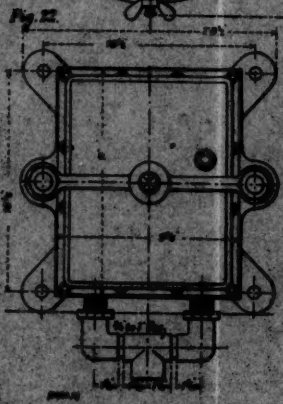


Fig. 22.

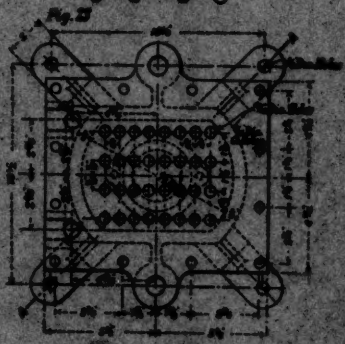
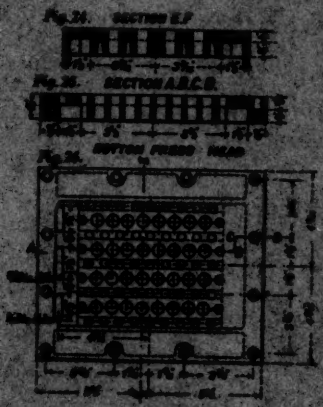
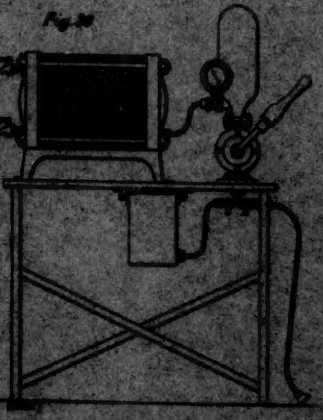


Fig. 23.

HELE-SHAW FILTER.





Ltd., Falmouth, to the design of the Gill Propeller Co. Ltd., 17, Victoria Street, London. A series of trials of the installation was carried out during the first week in July, and it was found that the speed of 11 knots could be attained with about 80 per cent of the indicated horsepower and 90 per cent of the revolutions per minute which were calculated as being required for an open screw. It was also found that the ship handled and steered with remarkable ease, even down to the working condition of just "turning over" at 12 R.P.M., while she could be stopped in 70 seconds from ringing down, when running at 10 knots, and showed notably rapid acceleration both "ahead" and "astern." It was also observed that the propeller displayed no tendency to race when breaking surface in heavy swell or to "cant" the ship when going from "ahead" to "astern." The performance of the propeller on this vessel is stated by those who witnessed the trials to be remarkable and conclusive, while it is reported to be equally satisfactory on the emergency service in which the vessel has been subsequently engaged, a speed of 11.5 knots having been attained.

It is also interesting to note that, when fitted to a large steam tug on the Thames, the propeller produced towing and steering results admittedly 5 per cent better than those attained with the best open propeller, in addition to eliminating the troubles of damage to blades, while when fitted this year to a large motor-driven canal tug at Liverpool, it gave a notably better towing speed than four open propellers of the best makes, previously tried on the same boat, and also proved immune from fouling troubles.

MANEUVERING POWERS.

The stopping and maneuvering powers developed by the Gill propeller appear to be quite good, and a ship so fitted answers her helm readily down to the lowest speeds. It has also been found to obviate squatting action at the stern and to leave a relatively smooth propeller wake.

The latter should prove a valuable attribute in the case of vessels operating in narrow rivers and canals. The employment of this type of shrouded propeller is not confined to any one class or size of vessel, for it is upheld that its range of utility is considerably wider than that of the open screw.

Its adoption in the case of vessels previously fitted with an open screw or screws does not involve any structural alteration or additional fittings to the hull or power plant, whilst its inclusion for a vessel in the design stage should permit of higher revolution speeds being employed with resulting economy in weight, bulk and cost of propelling machinery.—"Marine Engineer and Naval Architect," September, 1923.

STREAM LINE FILTERS AT THE OLYMPIA EXHIBITION.

Constructed by the Stream Line Filter Company, Limited, Engineers, London.

The Stream Line Filter Company, Limited, of 64, Victoria-street, London, are showing various types of filters constructed on the principle invented by Dr. Hele-Shaw, which consists in forcing the liquid to be filtered between the surfaces of closely packed sheets of paper. The results obtained by this method are really extraordinary. Not only can grosser impurities such as suspended matter be abstracted from a liquid, but emulsions can be separated into their constituents and even liquids colored by some substance in solution can be filtered clear and colorless. Moorland water, stained by peat, which has hitherto resisted every attempt to remove its color by filtration, can be delivered in a perfectly colorless condition, and even more striking to the ordinary observer is the total removal of color from a bril-

liant dye by its passage through the filter. The essential portion of the Stream Line filter consists of a stack of sheets of water-proof paper, pressed into contact with each other by suitable means. When so arranged, the numerous circular holes with which the sheets are perforated, coincide and form passages running from end to end of the stack. Of these passages about half the number communicate with the receiving end of the filter and the remainder with the delivery end. The liquid to be filtered is led under pressure to the interior of alternate passages and escapes by the remaining passages after passing between the faces of the water-proof paper sheets. The fineness of the passages through which the liquid has to pass can be adjusted by varying the pressure on the stack of paper. The impurities, or the matter to be eliminated, is retained round the walls of the inlet passages, whence it can be washed out or removed by a positive cleaning device.

Figs. 20 to 29 show the arrangement and the principal details of a Stream Line filter capable of dealing with about 250 gallons of liquid per hour. The paper is compressed between a pair of perforated plates, the upper of which is forced down to the required pressure by the screw and handwheel shown. The liquid to be filtered is led by the pipes shown to a grid-iron arrangement of channels in the upper plate. These channels are closed on their upper side by a rubber joint backed by a thin steel plate and held in position by the pressure of the head-casting. Referring to Fig. 23, which illustrates the top plate separately, it will be seen that the liquid passes down from the channels through four rows of circular holes, and so into the corresponding holes through the stack of paper. The lower plate is perforated with similar holes allowing the liquid under pressure to pass freely right through to the closed sludge tank below. The only escape for the liquid is by passing between the leaves of paper into rows of smaller holes through the paper stack. The position of these outlet holes is shown by the smaller circles seen in the plan view of the bottom plate, illustrated in Fig. 26. The small holes in the bottom plate do not go right through, but communicate with channels cut in the underside, where the filtrate collects and whence it is withdrawn by the pipe shown in Fig. 21. The material filtered out is retained inside the inlet holes through the paper stack and can be washed out by opening the sludge chamber at the bottom.

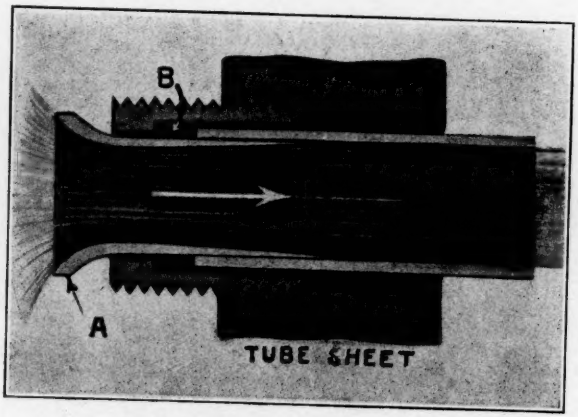
For large filters an improvement on this method has been devised, whereby the holes are positively cleaned out by the passage of a small plug, driven through them by the water pressure. In many commercial processes the important product of the filter is not the filtrate, but the matter filtered out, and in the Stream Line filter either or both can be recovered at will. The filter is made in small sizes for laboratory and experimental work, and a simple type, illustrated in Fig. 30, has been developed for household water supplies. On the stand is a model of a very large design in which hydraulic pressure is applied to the paper filtering material.—"Engineering," August 31, 1923.

CONDENSER TUBE CORROSION.

By R. Le Boysson,
Ingénieur principal de la marine française.

DISCUSSION OF RESULTS.

The English and Swiss experiments lead to rather different conceptions of the corrosion of condensers. The Swiss experiments being carried out with considerable care and precision appear to be the more comprehensive, and from them one can conclude that corrosion is not a chemical phenomenon



VENTURI NOZZLE PLACED AT INLET END OF TUBE.

but one of local electrolysis. Consequently the recommendations of Bengough can only be considered as palliatives. Only the correction of alkalinity can give a complete solution; and it remains to be seen whether it can be realized in a practical manner. However, the points upon which English and Swiss experiments are agreed ought to be borne in mind. Certain details should be considered; for instance, the influences of the galvanic effects of the condensate and that of the temperature. Experience has shown, in general, that auxiliary condensers which are generally rather hot do not resist corrosion well. It has been observed that condensers kept in reserve aboard ship, but not dry, suffer very quickly from corrosion, the cause being undoubtedly the alkalinity of the stagnant water.

The observance of the following rules is suggested therefore:—

1. *Nature of Metal.*—The condensers should be constructed throughout of the same metal as the tubes. The best metal in ordinary cases is brass (Cu 70, Zn 29, Sn 1) exposed for three hours at a temperature of 350 degrees F. ± 25 degrees F. to an oxidizing atmosphere. The iron contents should not exceed 0.1 per cent. Under microscopic examination the metal should only show large α crystals.

2. *Stuffing Boxes.*—Stuffing boxes should be fitted with a soft metallic ring at the bottom to ensure electric contact between the tubes and the plates. The circulating water pipes may be an important source of corrosion and hitherto unexplained troubles may be due to this cause; it is thus of the greatest importance to have the condensers perfectly tight.*

3. *Circulation.*—The cooling surface should be liberally designed and the circulating water should only pass through the condenser in one direction without multiple circuits and with spring chucks to ensure a uniform distribution. The velocity of the circulating water should not exceed 2 m/sec. It is better to keep it between 1.3 and 1.5 m/sec. Air suction pipes for the pumps should be avoided and the water should be drained before entering the condenser. Condenser heating is always dangerous and should be guarded against. The water inlet and ejection valves should be so disposed as to avoid circulation in a closed circuit.† All waste matter should be kept out of the circulating water.—"Marine Engineer and Naval Architect," July, 1923.

ENTRANCE NOZZLE FOR CONDENSER TUBES.

Among the many factors affecting the life of condenser tubes, general corrosion is probably the most common and can usually be attributed to one or a combination of causes. It is now generally conceded that the entrained and dissolved air in the circulating water is responsible for much of the corrosion action and that the life of condenser tubes depends quite as much on the character of the water as on the properties of the tubes.

With a view to eliminating the corrosive action due to the gas pockets formed at the inlet end of the tubes (as at B) the nozzle A shown in the illustration has been developed. The nozzle, when placed in a tube as shown, tends to prevent the contraction of the jets of water as they enter the tube and thereby stop the liberation of oxygen at that point.

*The tightness of the condenser is also of course of fundamental importance for the good functioning of boilers and turbines. The question arises whether stuffing boxes should not be dispensed with and the ends of the tubes expanded instead. This has long been the practice in the French Navy in torpedo boats with condensers of beaded tubes with non-parallel end plates. The question could be taken up for straight tubes of parallel plates. If the condenser is of the same metal as the tubes the differences of expansion are small and could be taken up by the elasticity of the tubes which being very long could easily bend without any danger.

† This desideratum is generally realized when the ship is going ahead. It also takes place when moored and even when going astern, particularly with turbines where going astern necessitates the use of large quantities of steam.

The design of the nozzle is such as to produce a venturi action and to increase the flow of water through the tubes.

The nozzles (known as Flowrites) are made in sizes to fit standard-diameter tubes and when in place form a practically continuous inside wall and provide a smooth entrance to the tube. They are manufactured and distributed by the Albert E. Mace Co., 755 Boylston St., Boston, Mass.—"Power," October 9, 1923.

THE RAPID CORROSION OF CONDENSER TUBES.

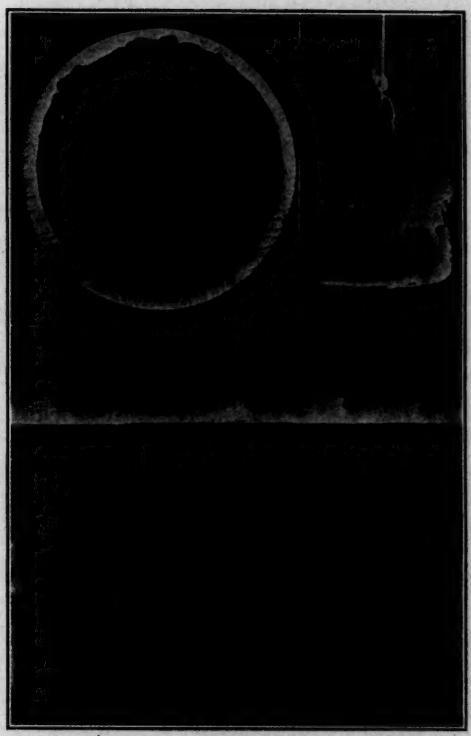
By Guy D. Bengough, M. A., D. Sc., and R. May, A. R. S. M.

This is a brief account of its authors' conclusions as to the cause of the rapid corrosion of condenser tubes which has sometimes occurred during the last few years.

It has been generally assumed in discussions on condenser practice that the speed of the water through any given tube in a condenser is approximately identical with the average speed through the whole condenser. This average speed is usually known, or, at any rate, it can be calculated with reasonable accuracy, but it has been found experimentally that the actual speed near the wall of any given tube may differ widely from the average speed. It is the speed at the tube wall that is important from the corrosion point of view. The matter has been investigated by the use of glass tubes of similar internal diameter to condenser tubes, and held in tube plates by ferrules of the ordinary design, the tube plates being attached to C.I. water ends in the usual way. A centrifugal pump was used to supply the water.

The phenomena of flow observed in the tubes varied considerably with the type of water used in the experiments. The behavior of sea water only will be discussed in the present article. The first point of importance to be noted is that smooth flow, such as is usually assumed to take place through all condenser tubes, did not take place through many of the tubes. In certain tubes the water stream formed vortices or more or less flat or extended spirals, especially for the first 2 feet or 3 feet of the tube. See Figs. 10 and 11. Where these spirals impinged on the walls local turbulence was set up—as shown in Fig. 11. Some approximate measurements were made—by a photographic method—on the variations of speed that occurred at various parts of the tube. For this purpose the introduction of small solid particles or air bubbles was useful; the behavior of these showed clearly that great variations of speed could take place even in a single tube. Thus at certain parts of a tube through which water was travelling at an average speed of 6 feet per second local retardation near the tube wall might reduce the speed almost to zero, while in the center of the tube the speed might approximate to twice the average speed. Turbulence prevailed in different degrees in many of the tubes for a length of 2 feet or 3 feet from the inlet end and then gradually died away. When turbulence prevailed high water speeds might be attained locally near the tube wall. The actual amount of water passing through a given tube was, of course, greatest in the tubes with least turbulence, *i.e.*, with the nearest approach to smooth flow. Probably the highest speeds were developed locally in tubes with the greatest turbulence. The phenomena differed noticeably in tubes in different positions relative to the main water inlet. Some tubes showed scarcely any vortices but local impingement on the tube wall near the inlet end—as shown in Fig. 9.

An increase in the water speed increased the complexity of the phenomena; a decrease to 2 feet or 3 feet per second caused an approximation to smooth flow to take place throughout the greater part of the tube.



FIGS. 3 TO 6.

It had previously been found experimentally that an increase in water speed, at least up to about 20 feet per second, increased the rate of corrosion of both copper and brass; consequently, the greatest local corrosion would be expected where the greatest turbulence prevailed, *i.e.*, near the inlet end.

Nevertheless, the failure of tubes in practice in a few weeks could not be explained on the basis of high water speed alone. Even with the highest

Fig. 8. Air bubbles shown black. Nearly smooth flow, characteristic of low water speeds. Large bubbles tend to rise to top of tube. Comparatively harmless form of flow.

Fig. 9. Air bubbles shown black. Bombardment of tube wall near end of tube. Produces heavy corrosion on top half of tube, and less corrosion on bottom half owing to the existence of eddies.

Fig. 10. Air bubbles shown black. Spiral motion without bombardment of tube wall. Comparatively harmless form of flow.

Fig. 11. Air bubbles shown black. Spiral motion and bombardment of tube wall. The worst condition for corrosion. The spirals "flicker" all round the tube and produce corrosion irregularly anywhere or all round the wall.

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FIGS. 8 TO 11.

water speed used in the experiments, namely, 25 feet per second, a copper tube would have a life of at least six months, at laboratory temperatures.

AIR THE CAUSE.

When working with sea water a phenomenon was encountered which was not observable with certain other types of water, namely, the sea water always contained a small amount of air in very fine bubbles. These were evidently due to a certain amount of churning in the centrifugal pump which had caused them to pass out of solution. They retain their identity for considerable periods in this particular water for the same reasons which cause it to exhibit the familiar property of foaming, which is not possessed by many fresh waters. The presence of these small bubbles was found to be useful in working out the movements of the sea water in the tubes, and additional air was added to the main water stream before it reached the water end by means of a needle valve which acted as an adjustable leak. Many striking phenomena were then observed with different arrangements of inlet pipe to tube. In particular, the bombardment of the tube wall with air bubbles at certain areas was noticed, and it seemed possible that such a bombardment might affect the rate of corrosion at such places.

Accordingly some comparative tests were made in which the corrosive effect of sea water containing only dissolved air in normal amounts was compared with that of the same water containing, in addition, mechanically entangled air. The results were very striking, and a few are shown in the following table. In the case of brass condenser tubes it will be seen that the rate of corrosion is greatly increased when a certain small volume of entangled air is exceeded; thus with 3 liters of air per hour passing through the tubes their lives would be only about three months in the cases shown in Table I. These tests were carried out by impinging a jet of sea water containing air on the surface of the tube arranged at right angles to it.*

TABLE I.—*Corrosion of Brass Tubes* (18 W.G.).

Time of experiment..... 28 days
Temperature..... 20 degrees C.
Speed of water ..... 10 feet per second

Alloy.	Per cent penetration on 1.21 mm. = 18 W.G.	Amount of entangled air in c.cm. per hour.
70:30	Less than 1	0.7
70:29:1*	2.5	0.7
70:30	Less than 1	25
70:30	Less than 1	145
70:30	31	3,000
70:29:1	35	3,000
70:30	83	11,000

* Laboratory cast specimen.

The total volume of air, however, is not the only factor in the case; another important one is its state of subdivision. A given volume of air is more harmful in a fine state of subdivision than when present in large bubbles. A fine foam appears to be the worst possible case. A foam consists of an intimate mixture of air and water, from which the air does not readily separate.

Very small amounts of certain liquids and solids can greatly stabilize foams, that is, prevent the separation of the air, *e.g.*, certain oils and basic salts. Notable differences in the tendency to foam have been encountered between samples of sea water collected in large stoneware jars and successively filled from the same position on a jetty; such differences may have a noticeable effect on corrosion. As a rough generalization it may be said, however, that the penetration of a tube per month by an air-sea-water jet containing three liters of air per hour in the conditions of Table I varies from 26 per cent to 33 per cent for the ordinary run of 70:30 tubes of commerce. With a highly foaming batch of water higher penetrations are occasionally observed, the highest yet recorded being 49 per cent. These experiments were usually carried out with sets of twelve tubes at a time, and it was clear that the varying results were due to the conditions rather than the tubes, since if one tube gave a high result all the others in that set did so. In all cases the type of corrosion strikingly resembled that which occurs in conditions of ordinary practice.

A small amount of sulphuretted hydrogen in the water, such as is occasionally found in conditions of practice, greatly enhances the rate of attack; thus, with sea water containing 3 cubic centimeters of the gas per liter, penetrations of 82 per cent and 90 per cent per month of twenty-eight days have been recorded in the conditions of Table I.

It is not proposed to discuss in detail in the present paper the exact way

*Varying the angle down to 45 degrees had but little effect on the result.

in which the presence of entangled air increases the rate of attack; from a purely chemical point of view any very large increase would appear to be improbable, since it is the air in solution that is supposed to be the active chemical agent. It may be said, however, that the action is largely physical and has to do with the penetration and eventual removal of a scale on the metal which is normally protective; under certain conditions of aeration no effective scale is formed. For instance, there is a tendency of basic salts, and possibly of other corrosion products to arrange themselves round the air bubbles, *i.e.*, at the air-water surface instead of at the metal-water interface, and this must clearly interfere with scale formation on the metal. A reason why small bubbles are more harmful than larger ones is that the former tend to be retained in small eddies near the tube wall, especially in the neighborhood of any obstruction, whereas larger bubbles have a greater tendency to travel along in the center or on the top of the stream. The presence of small bubbles at the seat of corrosive action not only increases the rate of certain chemical changes, particularly that from cuprous chloride to cupric chloride and oxychloride (atacamite), but also prevents the formation of a protective film as indicated above. Accordingly, one of the most remarkable characteristics of this type of action is the freedom of the corroded surface from corrosion products.

A high water speed increases the rate of action by forcing fine air bubbles into or through the normally protective scale.

Certain alloys, when subjected to the air-water jet test, have their surfaces beautifully etched. They become smooth enough and sufficiently free from corrosion products for microscopic study of the crystalline structure at moderate magnifications; others appear to have an almost polished appearance, especially those which have a "flowed" structure throughout the alloy. At first sight such alloys may seem to have been eroded mechanically rather than corroded, but there is much evidence against this view; the erosive action is confined to the products of corrosion only.

The reason for the peculiar distribution of the corrosion on tubes suffering from the rapid type of action described at the beginning of this article may now be suggested. It will be remembered that the position at which the action is usually worst is the first two or three feet of the tube at the inlet end. It is just at this position that turbulence and impingement of air on the tube wall are worst; here the tube suffers from an intermittent bombardment with air bubbles, as shown diagrammatically in Figs. 9 and 11. Further along the tube the turbulence dies out and corrosion falls off. The remarkable observation has been made, however, that the size of the bubbles gradually decreases, and towards the end of a long tube—*e.g.*, 15 feet—much of the air appears to be distributed as a fine "mist" of air in water, *i.e.*, in just the state in which it is most harmful. So fine is this mist that with an average speed of 7.5 feet per second through the tube no tendency for the bubbles to rise to the top of the tube could be seen. Any roughening of the tube surface by local scale accumulation or otherwise, or any obstruction in the tube will be liable to cause local eddies and local air bombardment, and consequently give a chance for corrosion to start. The matter is illustrated in Figs. 12 to 15.

Enough has now been said to show that if air is present in fine bubbles or as a foam in a rapidly flowing stream of sea water the rate of corrosion will be greatly increased as compared with similar stagnant water or water flowing at the same speed but free from entangled air. The question now arises as to whether such air is actually present in condenser circulating water in ordinary conditions of practice, and if so, how it gets there.

Unfortunately, in many cases, especially in marine practice, it has not been possible to determine by actual inspection whether or not such air was present. In a few cases in power station practice it has been possible to

observe the outlet water stream from the condensers; in all cases so far examined there has been a close correspondence between the speed and type of corrosion of new tubes and the presence of entangled air.

As regards the source of entangled air in circulating water much will depend on the particular conditions of a plant. In marine practice it may be suggested that entangled air would normally be present whenever the ship was in motion, and especially in rough weather. Leaks in the pipe line are probably of comparatively small importance, as the inlet train would

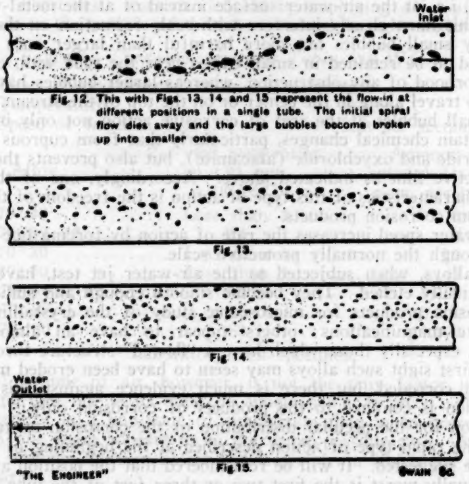


Fig. 12. This with Figs. 13, 14 and 15 represent the flow in at different positions in a single tube. The initial spiral flow dies away and the large bubbles become broken up into smaller ones.

Fig. 13.

Fig. 14.

Fig. 15.

FIGS. 12 TO 15.

probably be at or about atmospheric pressure. In many land plants, on the other hand, the condensers are frequently near the top of a siphon, the circulating water being sucked up many feet. In such cases a considerable portion of the inlet pipe train will be below atmospheric pressure and air will be drawn in at any loose joint. Thus the presence of a loose bolt and nut may cause corrosion of a new tube. Air leaks of considerable size can be detected by the accompanying hissing sound or by the use of a lighted taper, but small leaks may pass unnoticed for long periods and cause much trouble. The actual arrangement of the pipe trains and the tightness or otherwise of valves and gates will sometimes account for the irregular distribution of corrosion amongst the condensers of a single plant.

If it be true that entangled air is usually present in marine conditions, and often in power station practice, it would seem at first sight that rapid corrosion and failure of tubes should be much more common than it is. Even in modern conditions, however, serious corrosion troubles are only met with in a small proportion of marine and power station plants, though when they occur they may be very serious. Moreover, severe corrosion usually occurs in the early life of a tube, whereas entangled air may possibly be present during the greater part or even the whole of its life. How is it, then, that so many tubes successfully stand up to conditions that would appear to be corrosive on the views given above?

One reason for this is that entangled air often passes by preference through a comparatively small number of the tubes in a condenser, and the swirling in the water ends will probably decide which. Such swirling will be determined partly by the arrangement of inlet piping and water ends, and partly by obstructions such as steel protective blocks, &c., in the water ends (*cf.* Fig. 7).

This reasoning, however, does not account for the special liability of tubes to fail when new; for this a totally different explanation must be sought.

The explanation is that tubes can, in certain conditions, coat themselves with a scale consisting largely of corrosion products which will resist the attack of an air water stream. This suggestion is in harmony with a great number of observations that have been made on the general nature of corrosive action on copper and its alloys. Thus the rate of corrosion in stagnant water has been found to fall off with time, a fact that is readily explained by the gradual accumulation of scale on the metal. In fact, the conclusion has been drawn from a large amount of experimental work that the chemical stability of most of the common industrial metals and alloys in water is mainly dependent on the protective properties of the scale which forms on them, and that local pitting of the metal is due to local failure of the protective properties of the scale. Such failures may be brought about by either chemical or mechanical causes, or by both together.

If a tube of Admiralty alloy of 70:30 brass be kept immersed in stagnant sea water at 40 degrees C. for a week and then be removed, dried, and submitted to the air water jet test it will be found that its resistance has been greatly raised, as compared with that of the same tube tested without the preliminary treatment. Some comparative figures are shown in Table II Further tests on tubes, the scales of which had been dried previous to test, confirmed these results. The explanation of the failure of new tubes now becomes obvious. If the air water bombardment is to be strongly corrosive it must take place on metal entirely or nearly free from a sea scale. The formation and removal of the corrosion products must be nearly simultaneous; scale formation must not get decidedly ahead of scale removal if very rapid action is to take place. A sea scale, however, does not necessarily protect a faulty tube.

No doubt in time an air water attack will work through a scale formed on a tube in ordinary conditions of practice and cause corrosion beneath it, but the action, though more rapid than that of an unaerated water stream, will not be nearly so rapid as that on a new tube since there is always an intervening layer of scale hindering the action, and this layer once formed fairly thickly does not appear to be removed completely. Consequently, if during the early life of a tube in a condenser water is only slowly circulated and contains little or no entangled air, scale formation will get ahead of scale erosion and the tube may have a long life. In ships' condensers a slow circulation of water while in dock or a builder's yard when there is no motion of the ship may build up a good protective scale.

The experiments on scales described above were mainly carried out with scales formed at a temperature of 40 degrees C. It has been found that such a scale is decidedly more protective than one formed at ordinary atmospheric temperatures, though the latter has some protective value. It is interesting to note that as far as the corrosion of 70 : 30 and Admiralty brass tubes are concerned the increase of vacuum—and corresponding lowering of temperature—which has occurred in modern condenser practice has had a decidedly adverse effect on corrosion. The contrary is true of arsenical copper tubes; such tubes are more rapidly corroded at 50 degrees C. than at 20 degrees C.

TABLE II.

Time of experiment.....28 days
Temperature .....20 degrees C.
Speed of water.....10 ft. per second
All tubes tested without drying of scale previous to test.

Alloy.	Maker.	Per cent penetration on 1.21 cm. 18 W. G.	Treatment before test.
70 : 29 : 1	A	Less than 1	One week in stagnant sea water at 20 degrees C.
70 : 28.5 : 1.5	A	4	One week in stagnant sea water at 20 degrees C.
70 : 30	B	8	One week in stagnant sea water at 20 degrees C.
70 : 30	C	36	One week in stagnant sea water at 20 degrees C.
70 : 30	Average of six makers	30*	Untreated.
70 : 29 : 1	A	35	Untreated.
70 : 29 : 1	A	Less than 1	One week in stagnant sea water at 40 degrees C.
70 : 28.5 : 1.5	A	2.5	One week in stagnant sea water at 40 degrees C.
70 : 30	B	2 5	One week in stagnant sea water at 40 degrees C.
70 : 30	C	43	One week in stagnant sea water at 40 degrees C.
70 : 30	Average of six makers	30*	Untreated.
70 : 29 : 1	A	35	Untreated.

*Extreme figures were 26 and 33.

The complete failure of tube by maker C will not be discussed in this article, as it is due to an unusual cause; the result is included in the table to show that the conditions of test were actually severe and would attack a faulty tube.

PREVENTION.

It is now necessary to consider what measures can be taken to prevent the rapid type of corrosion described in this paper. On the general principle that prevention is better than cure, the simplest plan would seem to be to eliminate the entangled air from the circulating water. This is, no doubt, a simpler matter than the removal of the dissolved air, but in view of the immense volumes of water used in condensers it presents many difficulties. In marine conditions it would seem to be almost impossible. In power station practice something can be done to prevent air from becoming entangled in the water by attention to leaks in the pipe lines where they are below atmospheric pressure, and other measures may suggest themselves as locally suitable. In this connection it would be an advantage if means could be provided for the inspection of the outlet water so that the presence of air could be definitely ascertained.

If air once gets into the water stream it is very difficult to eliminate it by a settling process. The large bubbles can easily be removed, but the small bubbles only leave the stream slowly, and these are just the bubbles that are so harmful. Effective removal of these would probably require a large and costly apparatus, which would be commercially impracticable.

It would be a great advantage if records were kept of the *exact* positions and times of all tube failures in condensers. For this purpose it is convenient to use a large plan of the tube plate and mark off each tube on it when it fails. In many cases this would delimit the bad areas and prevent unnecessary re-tubing of whole condensers—a procedure occasionally carried out after a few failures on the assumption that the tubes are at fault and would, therefore, fail throughout the condenser. The authors' experience shows that many good tubes have been unnecessarily scrapped in this way. Remedial measures need only be applied to the affected areas in such cases; the rest of the tubes can safely be left in the condenser.

A simple method of reducing the trouble has already been suggested, namely, to use for replacements Admiralty tubes that have been heated to 40 degrees C. for a week in sea water, which should be stagnant or only slowly moving. If this cannot be done then tubes from other parts of the condenser in which failures are known to be few should be used for replacements, and new tubes inserted in the positions thus vacated. This procedure will often require tubes to be moved from the second pass to the first in two-pass condensers, and will be most effective if proper records of failures are kept.

It is perhaps desirable to emphasise that only one type of corrosion has been discussed in this article. Other types, quite different in origin, and requiring different remedial measures, also occur in practice. As an instance, so-called "dezincification" may be noted. Ten years ago the latter was the more important type, but owing to the increase in condensing efficiency with the consequent lowering of condenser temperatures and the increase of water speed, it is now of less importance than the type discussed in this article.

The work described in this article has been carried out by funds supplied by the Department of Industrial and Scientific Research, the British Electrical and Allied Manufacturers' Association, and the Brass and Copper Tube Association, to all of whom the committee's thanks are due.—"The Engineer," July 6, 1923.

NEW INFORMATION ON THE FATIGUE OF METALS

Since 1919 the Engineering Experiment Station of the University of Illinois has carried on, in co-operation with the National Research Council and the General Electric Co. an extensive series of experiments on the fatigue of metals. The first report, contained in Bulletin No. 124 of the Experiment Station, was abstracted in the December 13, 1921, issue of "Power." The work done since that time is described in a new bulletin, No. 136, which contains nearly one hundred pages of test results and conclusions of the greatest value to manufacturers of machinery in which rotating or other moving parts are subjected to repeated changes in stress.

Among the conclusions drawn from the work to date the following are of outstanding importance:

It has been fairly well established that ferrous wrought metals have a definite endurance limit, the endurance limit being defined as the unit stress below which a metal is capable of withstanding an indefinitely large number of complete reversals of stress.

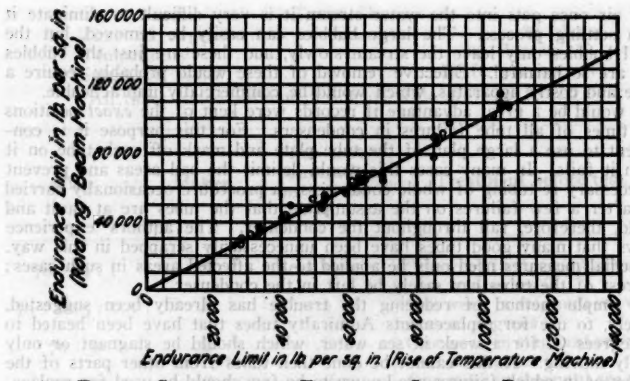


FIG. 1.—CORRELATION OF ENDURANCE LIMIT BY ROTATING-BEAM TESTS AND BY RISE-OF-TEMPERATURE TESTS.

This endurance limit may be determined with a fair degree of accuracy by a short-time test in which the rise of temperature under reversed stress is measured.

The endurance limit seems to be correlated with the ultimate tensile

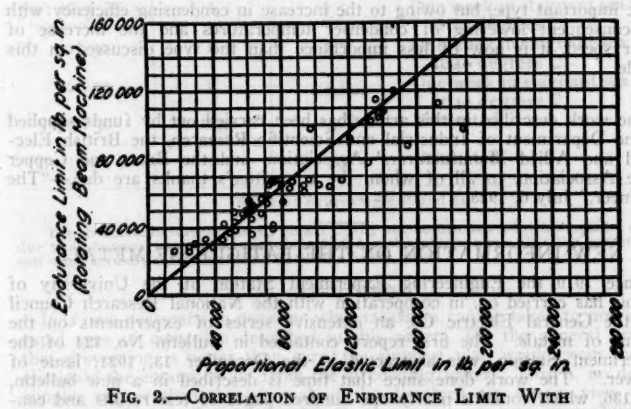


FIG. 2.—CORRELATION OF ENDURANCE LIMIT WITH PROPORTIONAL ELASTIC LIMIT.

strength, with the Brinell hardness number, and in a much smaller degree with the yield point and the proportional elastic limit. No correlation was found between the endurance limit and the ductility, the result of the Charpy impact tests of notched bars, or repeated impact tests.

The effect of speed of reversal of stress on the determination of the endurance limit seems to be slight.

The results of the tests of specimens with various temperatures of "draw," after oil quenching, indicate that for carbon-steel specimens tested, neither the ultimate tensile strength, the endurance limit for flexure nor the ductibility was appreciably affected by draws at a temperature lower than 600 degrees F. For nickel-steel specimens for draws up to 400 degrees F. the ultimate tensile strength and the endurance limit for flexure diminished slightly, while the ductility increased a little. For draws at higher temperatures the changes in values of strength and ductility were more marked, the ultimate tensile strength and endurance limit decreasing, and the ductility increasing. Whatever advantages may be gained by draws below about 600 degrees F., such as relief from internal stress and increased machinability, may be obtained with but little, if any, sacrifice of tensile strength or of endurance under flexure.

The results obtained with specimens subjected to severe tensile overstress before being tested in reversed bending indicate that a few applications of stress, well above the proportional elastic limit of the metal, lowered the endurance limit under subsequent reversed stress, the endurance being diminished by 22.9 per cent of its original value for one set of specimens.

If a specimen or machine part is subjected to cycles of repeated bending stress higher than the proportional elastic limit of the metal, the tests indicate that there may result a failure by excessive distortion of the specimen or the machine part. The endurance limit under repeated bending stress should, then, never be considered as higher than the proportional elastic limit of the metal.

Fig. 1 shows the close correlation that exists between the endurance limit as determined by the rotating-beam machine used in the test, and the same limit as determined quickly by the rise-of-temperature machine. While the correlation between the endurance limit and the other properties is not so

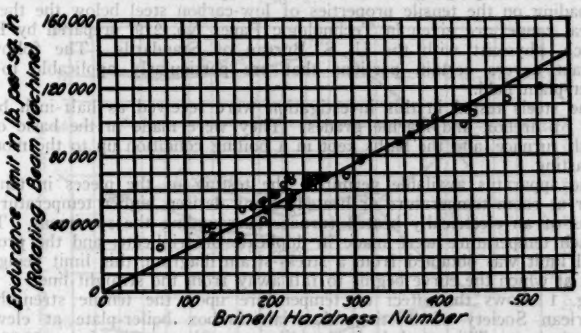


FIG. 3.—CORRELATION OF ENDURANCE LIMIT WITH BRINELL HARDNESS.

close, Figs. 2 and 3 show that there is a fairly definite relation between the endurance limit and the elastic limit and Brinell hardness respectively. Similar curves can be drawn for the yield point and ultimate tensile strength. —"Power," July 17, 1923.

ELECTRIC STEEL AT HIGH TEMPERATURE.

It is generally recognized that electric steel, that is, steel produced in the electric furnace, possesses many properties which make it superior to the steels produced by the ordinary methods, but while its use is being

gradually extended, its higher cost mitigates against its more general adoption. One of the most remarkable features of electric steel is its strength at high temperatures. From a series of tensile tests carried out in America over a range of temperatures from 60 degrees F. up to 1,200 degrees F., it was found that the strength of electric cast steel at 900 degrees F., which may be taken as red heat, is approximately 50,000 pounds per square inch, with a yield point of 30,000 pounds per square inch, while even at 1,200 degrees F. its strength is 23,000 pounds per square inch. The curves plotted from the results obtained show that the strength which is 79,000 pounds per square inch at 60 degrees F. gradually decreases to 70,000 pounds per square inch at a temperature of about 320 degrees F., and then increases again to about 76,000 pounds per square inch, which is reached at a temperature of about 550 degrees F. The strength then decreases to the values given above, following approximately a straight line law. The curve showing the yield point is of approximately the same form as that for strength, while the percentage elongation reaches a minimum at a temperature of about 600 degrees F., but a higher temperature does not appear to follow any definite law. It is clearly demonstrated, however, that within the usual range of temperatures for superheated steam, *i.e.*, up to 600 degrees F., when its strength is 69,000 pounds per square inch, electric cast steel possesses very great strength, which suggests that it is particularly well suited for the construction of the parts of high-pressure steam turbines.—"Shipbuilding and Shipping Record," September 13, 1923.

EFFECT OF TEMPERATURE ON PROPERTIES OF STEEL.

Some valuable findings on the effect of temperature, deformation, and rate of loading on the tensile properties of low-carbon steel below the thermal critical range, are given in Technologic Paper No. 219, prepared by H. J. French, physicist with the U. S. Bureau of Standards. The following abstract covers certain portions that are particularly applicable to the power-plant field.

The steels tested in this investigation were received as half-inch boiler plate of firebox and marine grades. They were made in the basic open-hearth furnace, and the baths kept in a boiling condition up to the moment of casting.

The apparatus available permitted the testing of the pieces in tension, either at room temperature or heated to any desired higher temperature by means of an electrically heated furnace surrounding the specimens. Tests at each temperature were made in duplicate or triplicate, and the proportional limit was obtained from a stress-strain diagram, this limit being the point at which the curve begins to fall away from the straight line.

Fig. 1 shows the effect of temperature upon the tensile strength of American Society of Testing Materials firebox boiler-plate at elevated temperatures. This steel had a tensile strength running from 52,000 to 62,000 pounds per square inch, and had the following composition: Carbon, 0.19; manganese, 0.43; phosphorus, 0.020; sulphur, 0.031 per cent. It will be noticed that the average tensile strength starting around 59,000 pounds at 70 degrees F., decreased slightly up to 200 degrees and then increased considerably to a maximum of 66,000 pounds at 560 degrees. Beyond this there was a rapid fall to 42,000 pounds at 860 degrees.

The reduction in area, as might have been expected, varied at all times in the opposite direction from the tensile strength. The elongation in two inches followed the general course of the reduction in area. Probably more important than the tensile strength is the proportional limit, as shown by the lower curve. This reached its maximum at a lower temperature than

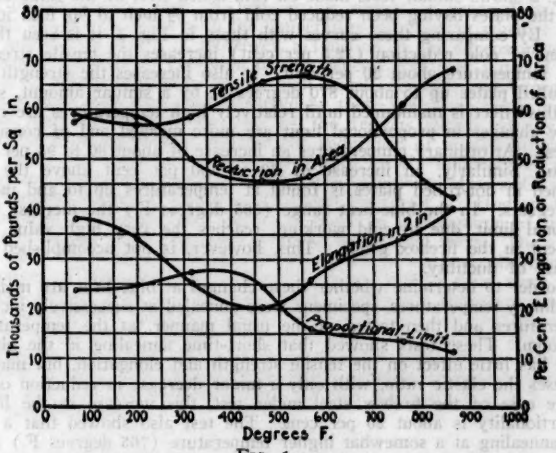


FIG. 1.

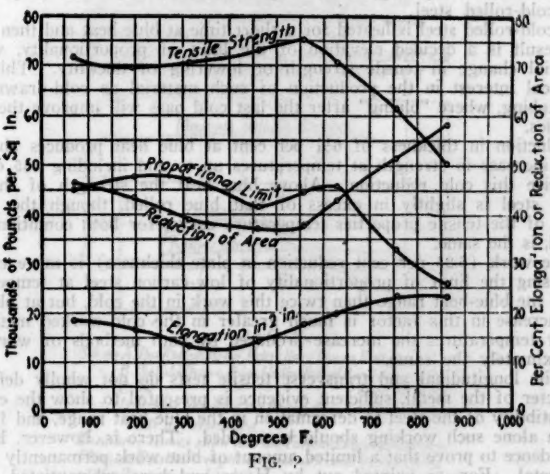


FIG. 2.

EFFECT OF TEMPERATURE ON PROPERTIES OF STEEL.

the tensile strength, and fell to 12,000 pounds at 850 degrees, as compared with 24,000 pounds at room temperature. These tests were made on the plate as rolled.

Fig. 2 shows similar tests made on cold-rolled steel of the same composition, the plates having been reduced cold from ½ inch to 7/16 inch in thickness. By comparing these curves with those in Fig. 1, it is seen that this amount of cold reduction (12.5 per cent) increases the tensile strength at room temperature about 20 per cent. It also increases the strength of the hot-rolled plates up to about 870 degrees F. by a similar amount, showing that this effect is maintained until relatively high temperatures are reached.

The changes in proportional limit are more marked and of considerable interest. At ordinary temperatures an increase of about 80 to 95 per cent is shown. Similarly, an increase of 60 to 100 per cent above the values obtained in hot-rolled plates is found at temperatures up to and including 470 degrees. In the blue-heat range (565 degrees F.) the increase in proportional limit, due to cold working, reaches the very high value of 200 per cent in the firebox grade. This, however, is not accomplished at the expense of ductility.

In order to determine whether these changes at blue heat are maintained at ordinary temperatures, specimens were annealed at successively increasing temperatures and then tested in the usual manner, at the temperature of the room. These tests showed that short-time annealing in the blue-heat range has little effect on the tensile strength and elongation, but materially increases the elastic ratio, with only a minor decrease in reduction of area. In the case of the firebox steel under test, this increase in the limit of proportionality is about 20 per cent. The test also showed that a short-time annealing at a somewhat higher temperature (765 degrees F.) accomplishes little, if anything, in the improvement of the tensile properties of this cold-rolled steel.

If cold-rolled steel is heated for a short time at blue heat and then cooled, the result is a decided elevation of the limit of proportionality, with no material change in tensile strength or lowering of ductility. This is of practical interest in the production of such material as cold-drawn light-wall tubing, where "bluing" after the last cold pass will improve the tensile qualities.

Reduction in thickness of 6¼ per cent at blue heat produces about the same increase in strength at temperatures up to and including 565 degrees, as twice this cold reduction. Above blue heat the strength of the cold-rolled steel is slightly in excess of that blue rolled, though the general shape of the tensile properties temperature curve, for both conditions mentioned, is the same.

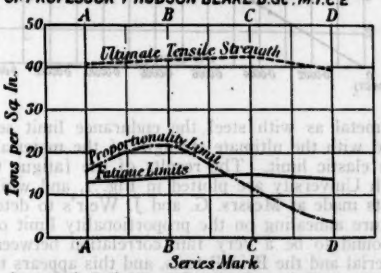
Blue work (6.25 per cent reduction in plate thickness) is more effective in raising the limit of proportionality of low-carbon steel at temperatures below the blue-heat range than twice this work in the cold, but at blue heat the increase in this factor is much greater in the cold-worked metal. At higher temperatures the increase produced by both methods of working is approximately the same.

While longitudinal and transverse tensile tests do not wholly define the character of the metal, sufficient evidence is presented to show the extreme susceptibility of the steel to deformation in the blue-heat range, and for that reason alone such working should be avoided. There is, however, little or no evidence to prove that a limited amount of blue work permanently injures the metal. For, as pointed out by Howe and here substantiated by the author, restoration of ductility may be obtained by annealing.—"Mechanical World," August 10, 1923.

THE FATIGUE LIMIT AND PROPORTIONALITY OF MONEL METAL.

The proportionality limit of non-ferrous metals is changed by cold-working, and it also alters with the temperature of finishing when it is hot rolled, and in this case it is, moreover, a function of the dimensions of the bar. Monel metal is no exception to this general rule, and some of those desirous of taking advantage of its high tensile strength and exceptional toughness have been somewhat disturbed by these variations in the proportionality limit, although in the case of the ferrous metals, at any rate, there is no close correlation between the endurance of a material and its elastic limit. In order to be in a position to satisfy inquirers as to the physical and mechanical significance of these variations, Messrs. G. and J. Weir, Limited, Cathcart, Glasgow, consulted Professor T. Hudson Beare, of Edinburgh University, who offered to make an investigation of the endurance of Monel metal specimens prepared in different ways. The experiments were made with a Haigh alternating stress machine, the materials being pieces of hot-rolled bar taken from stock; and to these were added specimens drawn cold to a circular form. Although the proportionality limit in the different specimens ranged from 4 tons to 21.2 tons per square inch, the investigation showed that the fatigue limit varied remarkably little, ranging only from 13½ tons to 15 tons per square inch. These results seem to show that there is even less correlation between the proportionality limit and the endurance limit for this material, than is found to be the case with steel.

Fig. 1. DIAGRAM SHOWING THE RESULTS OF FATIGUE TESTS ON MONEL METAL BARS MADE AT THE EDINBURGH UNIVERSITY BY Mr. W. GORDON B. Sc., M.I.M.E., UNDER THE DIRECTION OF PROFESSOR T HUDSON BEARE B.Sc. M.I.C.E



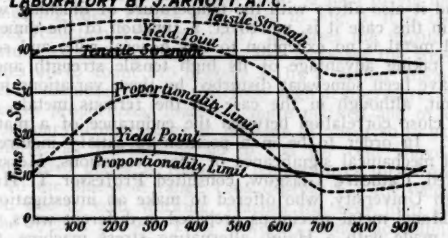
Series A. Hot Rolled Monel Metal Bar 1" Dia. as Rolled
" B. ⅞" Cold Drawn from 1" Hot Rolled Bar & Annealed
" C. ⅞" " " " " " " & not Annealed
" D. ½" Whit. Hex. Bar Cold Drawn from 1⅛" Hot Rolled Bar not Annealed.

The above Curves show how the Fatigue Limit follows the Ultimate Tensile Strength, & is not affected by Variations of the Proportionality Limit.

The Fatigue Limit Curve is compiled from Experiments which covered Reversals up to 16,848,000 on Unbroken Specimens.

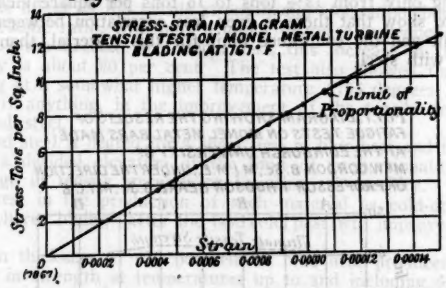
(7930 A)

Fig. 2. CURVES SHOWING THE RELATION BETWEEN THE ANNEALING TEMPERATURE & PROPORTIONALITY LIMIT, YIELD POINT, & TENSILE STRENGTH OF MONEL METAL RODS. TEST MADE AT MESSRS. G. & J. WEIR'S LABORATORY BY J. ARNOTT, A.I.C.



Dotted Lines. 5/8 Cold Drawn Rod. Full Lines. 3/4 Hot Rolled Rod.
(7930.8)

Fig. 3.


STRESS-STRAIN DIAGRAM
TENSILE TEST ON MONEL METAL TURBINE
BLADING AT 767° F
Limit of Proportionality
Stress-Tons per Sq. Inch.
Strain
0 2 4 6 8 10 12 14
0 0·0002 0·0004 0·0006 0·0008 0·0010 0·0012 0·0014

(7867)

With Monel metal as with steel the endurance limit seems to be very closely connected with the ultimate strength of the material, and to depend but little on the elastic limit. The results of the fatigue tests carried out at the Edinburgh University are plotted in Fig. 1, and whilst Fig. 2 shows the result of tests made at Messrs. G. and J. Weir's to determine the effect of low temperature annealing on the proportionality limit of Monel. With steel, there is found to be a very fair correlation between the endurance limit of the material and the Brinell tests, and this appears to hold also with Monel metal. Thus with a series of 9 specimens on which the proportionality limit ranged from 10.3 up to 29.2 tons per square inch, the range of Brinell numbers was from 163 up to 217. The lower figure is, however, unfairly low as the specimen in question had been heated to 800 degrees C. and rapidly cooled. This specimen had a proportionality limit of 18.4 tons per square inch.

For comparative purposes, therefore, it would be fairer to take the Brinell range as from 187 to 217, whilst, as stated, the proportionality limit ranged from 10.3 up to 29.2 tons. Messrs. Weir have also had some further experiments made at the National Physical Laboratory, with finished turbine

blades of Monel metal, in order to determine the variation of the elastic properties with temperature. The specimens had an area of 0.325 square inch, and were held in specially made grips. The results observed were as follows:

Temperature of specimen	Room Temperature	767 degrees F.
Area of specimen square inch	0.325	0.325
Limit of proportionality, tons per square inch	13.9	9.2
Yield stress, tons per square inch	37.7	31.2
Ultimate stress, tons per square inch	41.9	36.4
Modulus of elasticity	$23.9=10^6$	19.3×10^6
Elongation on 2.3 inches	—	16.5 p.c.

We reproduce in Fig. 3 annexed a graph which shows that even at this high temperature the elasticity of the metal is practically perfect up to 9.2 tons per square inch.—"Engineering," July 20, 1923.

BOOK REVIEWS.

ALFRED YARROW—HIS LIFE AND WORK, BY ELEANOR C. BARNES (LADY YARROW). Longmans, Green & Co., New York City.

A very intimate biography of one of the greatest of marine engineers. Owing to his close connection with the development of marine engineering, a history of his life must necessarily be a review of the accomplishments in that science. The human touch, however, gives an added interest to the book.

DIESEL AND OIL ENGINE HAND BOOK, BY JULIUS ROSBLOOM. The Technical Publishing Co., Los Angeles, Cal.

An elementary description of various types of marine engines; also of some of the principal auxiliaries used with Diesel engines; also of typical marine installations and the method of operating them. Of more interest to students than engineers.

OIL FLOW—VISCOSITY AND HEAT TRANSFER, BY R. S. DANFORTH. Published by R. S. Danforth, 525 Market St., San Francisco, Cal.

A very useful book for all designers of fuel oil apparatus. A great deal of information has been collected and is given in the form of easy reading charts.

BLUE PRINT READING, BY JOSEPH BRAHDY. D. Van Nostrand Co., New York, N. Y.

An elementary treatise on this subject for students.

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